STUDY OF A CLUTCH OPERATED POWER SCREW ACTUATOR FOR SPACE APPLICATION

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AN EVALUATION OF BASIC CLUTCH TYPES

PART 1: Mechanical Contact Clutches for Space Application, by R. I. Anderson

Submitted under separate cover:

PART 2: Electro-Magnetic-Flux Clutches for Space Application, by C. A. Depken

PART 3: An Evaluation of Hydrodynamic Fluid Clutches for Space Application, by W. A. Jones

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MECHANICAL-CONTACT CLUTCHES

FOR

SPACE APPLICATIONS

A THESIS

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by

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MECHANICAL-CONTACT CLUTCHES

FOR

SPACE APPLICATIONS

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SUMMARY

The object of the study reported herein was to determine the characteristics, capabilities, and limitations related to the applications of mechanical-contact clutches in space.

Ten clutches representing nine manufacturers were inspected and tested. Torque-to-weight and torque-to-volume ratios were considered, and actuating power requirements were compared. Tests made on these clutches included determining time requirements for engagement and for disengagement. To determine the effect of wear, one clutch was cycled over 50,000 times with average torque slightly greater than rated. This same clutch was heated in an oven to a steady state temperature of 165 F to determine the effect of high temperature on its characteristics.

A discussion of heat transfer considerations related to the application of clutches in space is presented.

Wrap-spring clutches were found to have larger torque-to-weight and torque-to-volume ratios than those of disc clutches. Spring clutches also required less power for actuation. Results of the testing showed that the time history of torque for all the clutches was in general the same. The wrap-spring was the fastest responding clutch, but problems were encountered due to high drag torques and unreliable disengagements. Neither wear nor high temperature appeared to seriously alter the clutch characteristics. Maximum torque was found to decrease slightly for the electromagnetically actuated clutch when it was operated at 165 F. However, the conservative rated torque of the clutch was still obtainable.

Forcing circuits were found very effective in improving the time response of electrically actuated clutches.

NOMENCLATURE

A	area, square feet
C	capacitance, farads
E	voltage, volts
e	constant, 2.71828 . . .
h_c	contact conductance, Btu/hr-ft^2-F
I	inertia, in-lbs-sec.^{2}
K	thermal conductivity, Btu/hr-ft-F
k	constant
L	inductance, henries
P	pressure, psi
Q	heat transfer, Btu/hr
Q_c	heat transfer by conduction, Btu/hr
Q_e	heat transfer from electric coil, Btu/hr
Q_s	heat transfer due to friction, Btu/hr
R	resistance, ohms; when referring to heat transfer, thermal resistance, hr-F/Btu
s	Laplace operator
T	torque, inch-pounds
T^o	temperature, degrees Fahrenheit (F)
$\bar{T}$	temperature, degrees Rankine (R)
t	time, seconds
W	work, inch-pounds
Y_o	yield stress, psi

ε	thermal emissivity
θ	position, radians
$\dot{\theta}$	velocity, $d\theta/dt$, radians/second
$\ddot{\theta}$	acceleration, $d^2\theta/dt^2$, radians/second^2
μ	coefficient of friction
σ	Stefan-Boltzmann constant, 0.1714×10^{-8} Btu/hr-ft^2-R^4
τ	time constant, seconds

CHAPTER I

INTRODUCTION AND HISTORICAL BACKGROUND

Mechanical-contact clutches have been used for many years as a means of controlled power transmission. The theory and the design of these clutches have not changed considerably, but the discovered applications are almost innumerable.

The first applications of the mechanical-contact clutch were in the design of industrial machinery. Since the invention of the automobile, this clutch has played an important role in the automotive industry, and with the development of the automatic control field in the past twenty years, the number of applications has soared. To meet the applications in the control field, many clutches have been designed to be actuated from a remote source or by a signal emitted by some piece of equipment in the system.

With such success as has been accomplished in the automatic control field, it is not surprising that clutches are being considered for application in the space effort. A clutch that can respond to signals from a remote source is required, and the controls field already has such a clutch -- many of them, in fact, with a variety of capacities and characteristics.

Objective

The object of the author's work has been to obtain the characteristics mentioned above and determine the capabilities and limitations related to the application of mechanical-contact clutches in space. To accomplish his object, the author has tested clutches manufactured presently for industrial and automatic control applications. It is important to realize that these clutches were not designed specifically, if at all, for use in space. However, by extrapolation of the data obtained, it is hoped that a clutch can be designed that will be applicable in the space effort.

Mechanical-Contact Clutches

Friction clutches are the oldest type of mechanical-contact clutches. Although there are a number of ways of applying the principle of friction clutches, this principle is very basic (1): If a body A is moving to the right (see Figure 1), and body B is pressed against A by a force F, and if the coefficient of friction of the mating surfaces is μ, then the maximum force that can be transmitted by frictional contact in the plane of the mating surfaces is

$$F_R = \mu F$$

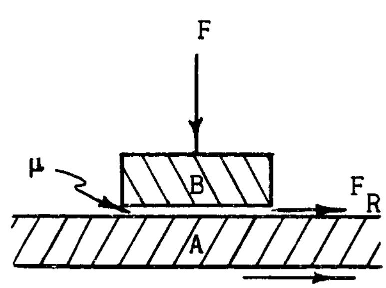

Figure 1. Basic Friction Principle.

The manner in which the contact is made -- whether it is simply two flat plates pressed together, or a drum that expands to come in contact with the shell that surrounds it, or a flexible shell that expands inward to make contact with the drum it surrounds -- constitutes the primary difference in friction clutches. Then each of these types may be classified by how they are actuated; mechanically, electrically, pneumatically, or hydraulically (2).

The wrap-spring clutch also has a simple principle. If a spring is twisted in the direction it is wound, the diameter of the spring becomes smaller, causing the spring to grip onto that around which it is wound. If the spring is twisted in the opposite direction, the diameter of the spring will become larger, releasing any gripping force. A spring wrapped around two shafts could connect or disconnect the shafts, thus controlling power transmission.

Usually, the spring of a wrap-spring clutch is wound so that its inside diameter is smaller than the diameter of both driving and driven ends, producing a radial force on the shafts. The principle of operation depends upon this radial force combined with the coefficient of friction between the spring and the shafts. To disengage the clutch, a tang connected to the end of the spring is held or turned opposite the direction of the turning clutch, thus unwinding the spring. A variation used by some manufacturers is to have an interference fit only on one end, and actuation is achieved by holding the other end against the shaft, wrapping the spring around it. Tapered (thickness) springs are sometimes used to reduce the inertia of the wrapping end.

Related Literature Survey

In 1924, R. Waring-Brown (3) wrote his book Friction Clutches in which he discussed the theory of the friction clutches of that time, which, in view of its simplicity, unsurprisingly is the same today. Since that time, the literature yields only articles on new applications and variations of the basic theory.

The automotive industry and the manufacturers of clutches are the major sources of data on the characteristics of mechanical-contact clutches. Before new clutches are manufactured, prototypes pass many tests to obtain such data as release load, total "actual" plate force, the torque capacity, and the burst limit r.p.m. (4). Some clutches are even tested to destruction (5).

Concern for the response time of clutches has been brought about with the development of the automatic control field. For example, the National Bureau of Standards used a clutch to advance the film in a camera that photographed randomly occurring events, the events automatically triggering the camera (6). A response time of less than seven milliseconds was required for the clutch. However, the response times of clutches, other than those specifically designed for automatic control, are not generally given in the manufacturer's specifications.

Spring clutches, although not as old as the friction type of clutch, have been used for years. The theory of the spring clutch and how to design it was presented in 1939 by Wiebusch (7), and then later by Kaplan (8, 9).

Advantages of the spring clutch have been found to be its fast response and its high ratio of torque to inertia. Leonard (10) speaks of spring clutches that can bring a rated load up to a full speed in less than two milliseconds. He does not, however, give specifics on how large the rated load is.

One disadvantage of the spring clutch is that it can only deliver torque in one direction, whereas most friction clutches deliver torque irregardless of the direction of rotation. Another disadvantage is the clutch's unpredictable backlash, although Kaplan (11) has eliminated this backlash by modifying the spring.

Actuation of clutches in space will be probably done from a source remote from the clutch. Electrical actuation and pneumatic or hydraulic actuation using a solenoid valve are the types frequently used. Of course, the total response time for actuation depends not only on the clutch but on the response time of the actuator as well.

Forcing circuits are usually used where fast response is desired. By applying a higher-than-rated voltage across the coil and then reducing it to the rated working voltage as the flux reaches its steady-state value, full torque is applied to the load much sooner, and engagement and disengagement times can be reduced correspondingly.

As illustrated by Saliatesta (12), some circuits are good only for fast response when engaging the clutch. They do not cause a quick release of the clutch. Figure 2 shows two such circuits. In circuit (a), both switches 1 and 2 are closed to actuate the unit, allowing a high voltage to be applied to the coil. After a time delay, switch 2 is opened reducing the voltage across the coil to the correct working voltage. Circuit (b) shows an R-C network where, upon closing the switch, an initial high voltage is supplied to the coil due to the capacitor discharge. The resistor, however, reduces the steady state voltage to the rated voltage for the coil.

Some circuits aid only in the release of the clutch. Figure 3 shows one where a capacitor is shunted with the clutch.

A vacuum tube circuit such as shown in Figure 4 can be used to speed up both energizing and de-energizing response times.

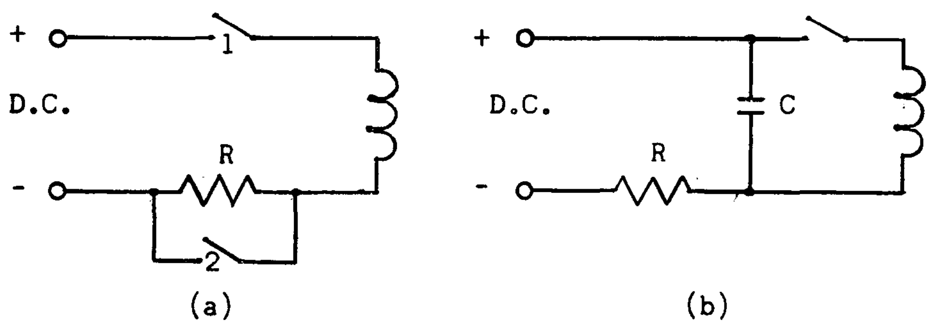

Figure 2. Quick-Engagement Circuits.

P. Block and D. Hennings, in their "Automatic Switching Control with Electric Clutches and Brakes" (13), describe a number of actuation circuits, some using tungsten lamp bulbs.

It is important to note that, as Block and Hennings point out (14), when an energized clutch circuit is opened, the collapse of the magnetic field produces a voltage surge that, unless suppressed, can produce severe arcing at the switch. A capacitor positioned as in Figure 3 is the most common method of suppression.

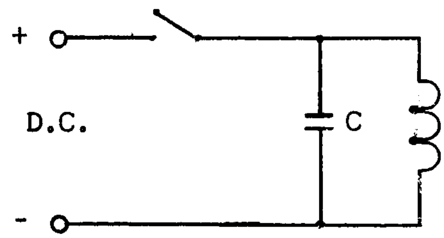

Figure 3. Quick-Release Circuit.

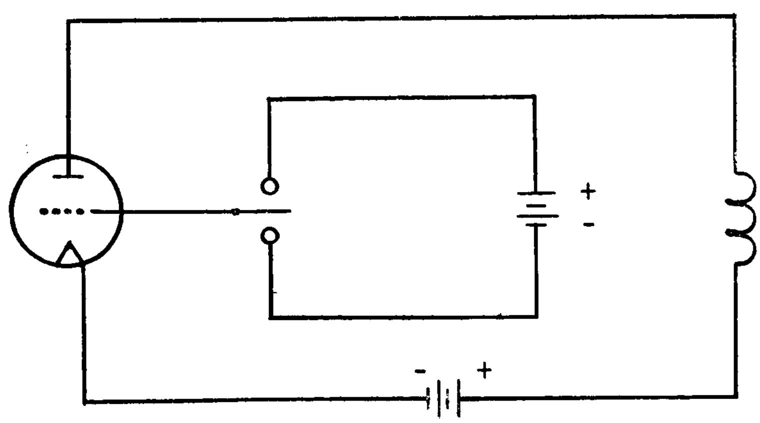

Figure 4. Vacuum Tube Circuit.

Pneumatic and hydraulic actuation can be made faster by placing the control valve as close to the clutch as possible, thus reducing the amount of fluid that is compressed when the clutch is actuated. Faster response of the solenoid valve can be obtained by using one of the forcing circuits mentioned above. Fast release may sometimes be accomplished by exhausting the fluid directly from the clutch to the surroundings rather than back through the control valve. In space, however, it may not be desirable to exhaust the fluid to the surroundings.

A number of industrial and design periodicals have in recent years carried summaries of available clutches and manufacturers. Most recent of these summaries are by Wardle (15), Harrison (16), Taylor (17), Gagne (18), and Botstiber (19). Annett (20), Barron (21), and Hyler (22) are authors of earlier articles. Most of these articles, however, omit specific data on performance and do not compare one clutch with another. Some of this information will be furnished in the work presented here.

CHAPTER II

EXPERIMENTAL INVESTIGATIONS

Purpose of Experimentation

To obtain data on torque characteristics, tests were made on ten commercially-available clutches, representing nine different clutch manufacturers.

Experimental Equipment

A block diagram of the test apparatus is shown in Figure 5. As can be seen from the diagram, the apparatus was a velocity control system using feedback.

For its stiffness, fast response, and variable speed capabilities, a hydraulic drive was used. Drive speed was obtained by using a photoelectric counter to count the teeth on a gear, and the count was displayed each second on an electronic counter.

The clutch to be tested was positioned after the speed control feedback loop. Following the clutch, a strain gage torque transducer was used to measure torque build-up. In order to load the clutches, a fixed end was provided with an adjustable slip clutch between it and the torque transducer. An inertia load could also be placed on the output shaft of the transducer and be used alone or in combination with the slip clutch.

A dual-trace storage oscilloscope was used to record the time history of torque, and a Polaroid camera was used for permanent recording.

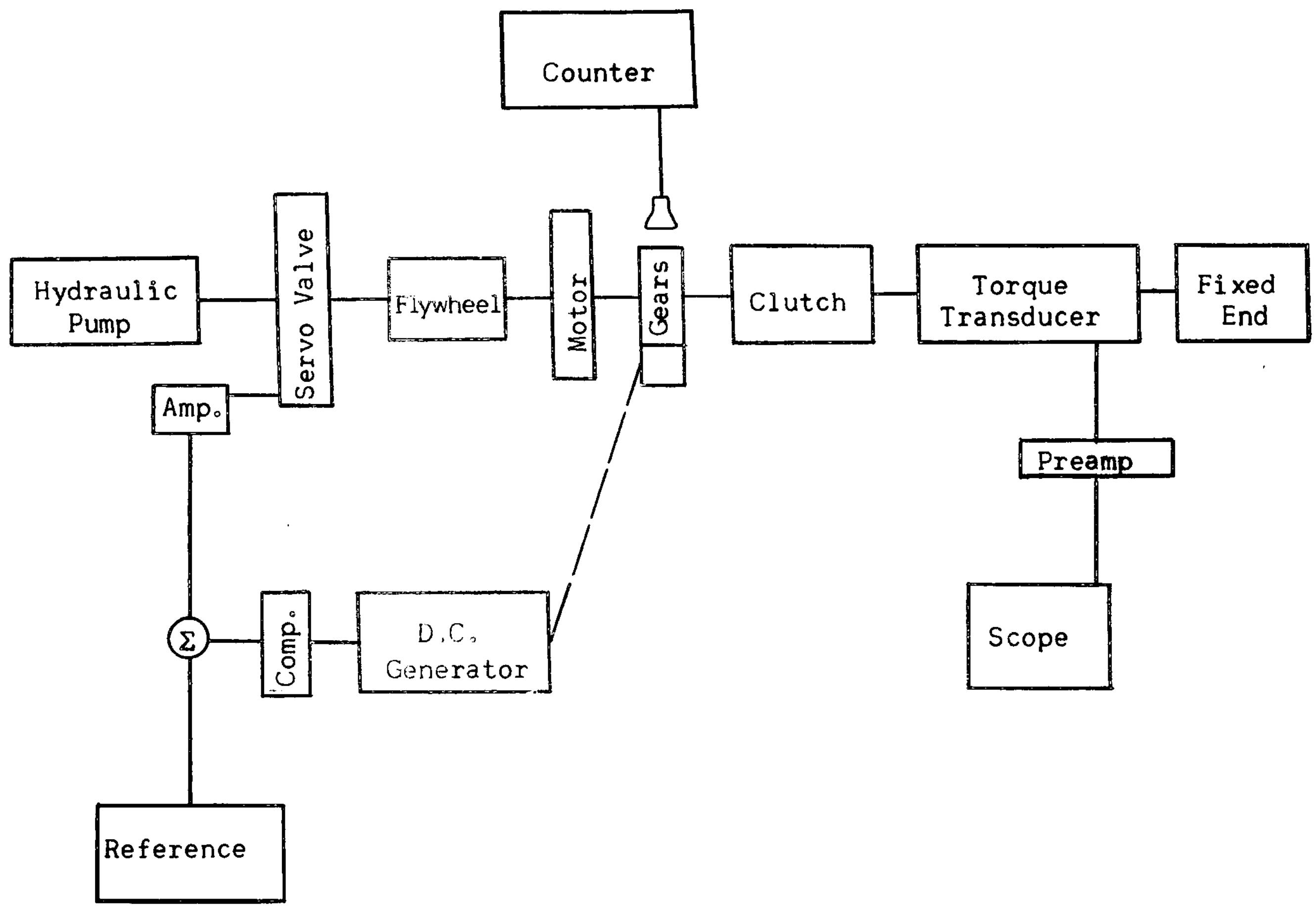

Figure 5. Block Diagram of Test Apparatus.

Most clutches were actuated electrically, thus the time history of actuating current was of interest. This history was plotted on the oscilloscope along with the torque history. Actually, the voltage across a known resistor in the clutch circuit was used to obtain the current history.

Pressure build-up of pneumatically actuated clutches was recorded on the oscilloscope by means of a pressure transducer.

Detailed description of the instrumentation used is given in Appendix A.

Experimental Procedure

Time responses of the clutches were of primary interest, so tests were oriented around time history of torque. Tests were run at various speeds, changing actuating voltage, or changing actuating pressure in the case of pneumatically actuated clutches.

The time history of current (pressure) build-up was recorded on the oscilloscope simultaneously with the torque build-up, so that later, curves could be plotted of torque versus current (pressure).

Tests on all electrically actuated clutches were run first without the use of forcing circuits. Additional tests were made on some of these clutches to check the effect of such circuits on the response times.

Plans were at first made to test torque build-up with two types of loads, inertia and friction. Upon further study, it was seen that the torque build-up of both loadings would be identical. Once the output of the clutch began moving in the case of a friction load, or once the inertia load was brought up to the clutch input speed (steady state velocity), the torques of the two systems would be different. But during the time of transient torque, which was of prime interest, the curves would be the same. This was demonstrated on the test apparatus, an example of which is shown in Figure 6. Thus the simpler setup, employing a fixed output with an interposed slip clutch, was used as the primary test apparatus.

One clutch was tested new, and was then cycled over 50,000 times and tested again. This same clutch was tested at a high temperature of 165 F. In the case of this particular clutch, which employed a special ceramic friction surface material, no significant changes in operating characteristics resulted.

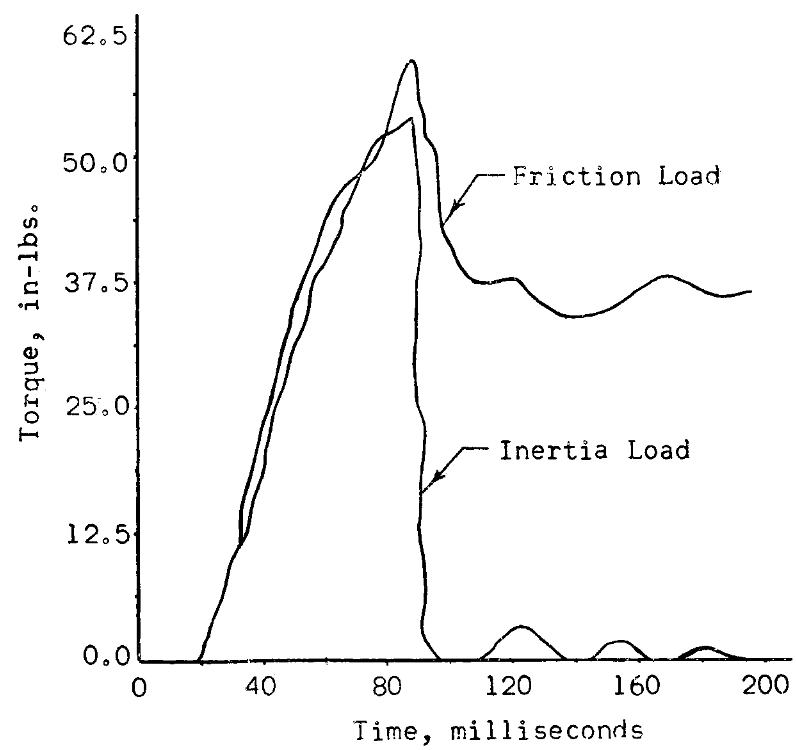

Figure 6. Comparison of Torque Build-up for Inertia and Friction Loads.

Torque-Time Relationships

Disc Clutches

A typical time history of torque of a disc clutch, electromagnetically actuated, is shown in Figure 7. The time history of current for this type of clutch is also plotted. Actual data obtained from the clutches of this type are shown in Appendix B, Figure 30 through Figure 41.

At $t = 0$, the switch controlling the clutch was closed and the solenoid actuated. For $0 < t < t_1$, although current was building up, there was no torque build-up. Disc clutches have a finite gap between their plates, and it takes some current going through the coil to cause the plates to move together. During t_1 units of time the flux density built up sufficiently to move the plates, and they actually closed the gap. At $t = t_1$ the plates made contact.

Actually only one plate usually moves. Some clutches use a stationary field and a segmented iron armature which mounts on drive pins of a spline to permit axial movement and which is connected to the output shaft. An intermediate rotor is used to magnetically couple the field to the armature. The rotor and the armature are the members of the clutch which upon contact transmit torque. The rotor does not physically contact the field. Some disc clutches have fields that revolve with the clutch. The armature can be connected to either the input or the output shaft. When the coil is energized through slip rings, the armature moves, and the friction force of one member on the other transmits torque. In both of the above clutches, the movement of the armature changes the inductance of the coil assembly and shows up in the time history of current as a "pip."

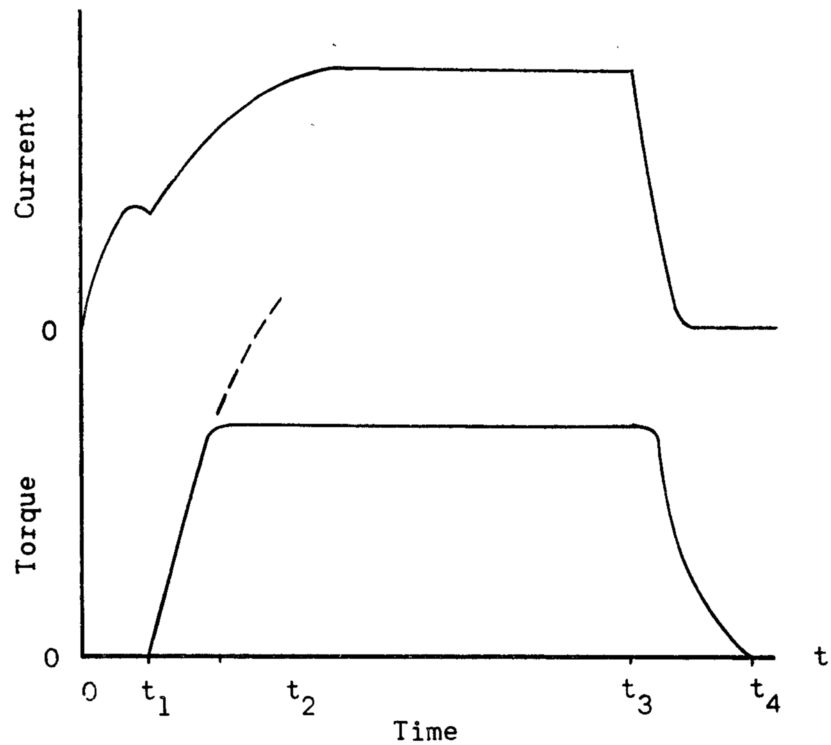

Figure 7. Torque and Current History of Electromagnetically Actuated Clutches.

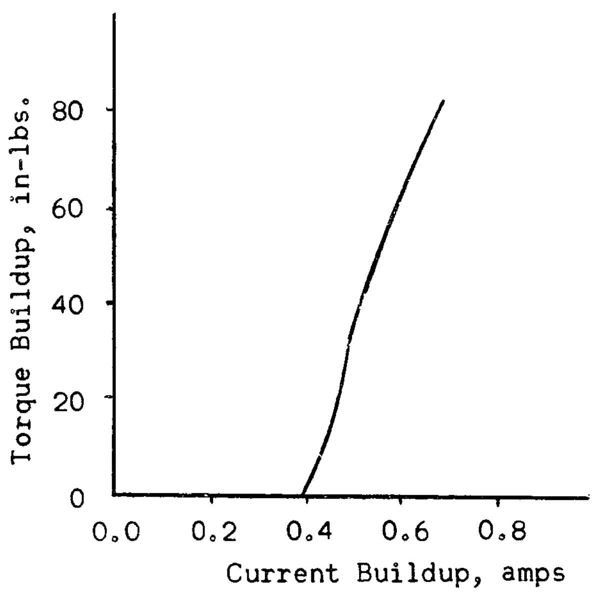

Figure 8. Torque vs. Current Build-ups of Clutch C.

During $t_1 < t < t_2$, torque built up to its maximum. It was found that for most disc clutches the torque build-up was nearly linear if the maximum torque applied to the clutch was within the range for which the clutch was rated. As the clutch reached its upper torque limit, the build-up slowed, making the curve have a form similar to

$$\text{Torque} = k_1(1 - e^{-k_2 t}) \tag{2.1}$$

where k_1 and k_2 are both constants. Because this is the same form as the current build-up in a coil, the curve of torque build-up versus current build-up could be a straight line if the resistance and inductance of the coil were properly chosen. Unfortunately, very seldom is this the case, but as a first approximation a straight-line relationship may be used. A curve of torque build-up versus current build-up for Clutch C is shown in Figure 8.

Due to the type of loading, i.e., a slip clutch preset at the desired torque, the torque curve peaked at the end of build-up and then dropped back to a steady state value (see Appendix B, Figure 31). This was because the static friction of the slip clutch was higher than its kinetic friction. Since this was characteristic of the load and not of the clutch being tested, the peak was not shown as part of the torque history for the clutch.

Referring back to Figure 7, for $t_2 < t < t_3$ the clutch turned the load, and output speed was the same as the input speed of the clutch, i.e., no slip. In steady state, friction clutches do not slip unless the clutch is loaded beyond its torque capacity.

At $t = t_3$, the switch to the clutch was opened. Current dissipated from the clutch coil very rapidly, but the torque did not decrease as fast. In fact, unless the torque was the maximum for the clutch, there was a time delay after the current began to dissipate before torque began to drop. The flux had to deteriorate to the point where it no longer applied enough force between the plates to maintain the required torque. Residual magnetism has the effect of holding the plates together even after the d.c. signal has been removed. Once started, the torque decreased in a smooth curve, and the dissipation time was usually in the order of 50% of the actuation response time (see Figure 30 in Appendix B).

Although not indicated in Figure 7, some of the data on the electromagnetically actuated clutches show discontinuities in the torque curves, occurring at the moment the switch was opened. This discontinuity was not a characteristic of the torque, but instead was due to the sudden arc across the switch when it opened. The arc was sometimes picked up electrically on the scope causing the discontinuity. Figure 7 was drawn without the discontinuity since the torque curve was not really disconnected at $t = t_3$.

Many clutches have springs to speed up the disengagement time. If a spring is used, the magnetic field of the clutch must be sufficiently strong to overcome the spring force and to yield good contact between the clutching surfaces. The spring, on the other hand, must be strong enough to overcome the holding action of residual magnetism. Usually a compromise is made when designing the spring (23). Note Figures 33 and 40 of Appendix B. Clutch E used a spring to help disengage, but Clutch B did not. The improvement due to the spring is significant.

Although the curve of torque build-up versus time for a pneumatically actuated disc clutch is similar to the one for an electromagnetically actuated clutch, the time history of pressure build-up is different than that of current as can be seen in Figure 9. Pressure build-up with respect to time is almost a straight line, and its slope is not as steep as that of an electrical build-up in a coil.

Explanation of the curves in Figure 9 is similar to the explanation for the curves obtained with electromagnetically actuated clutches. Air was controlled by an ASCO solenoid 3-way valve as mentioned before. Due to the solenoid there was a time delay between the actuating signal (switch closed) and the opening of the valve indicated on the curves by the pressure beginning to increase. For the system used, this delay amounted to about 9 milliseconds. The end of the delay is indicated in Figure 9 by t_1. From t_1 the pressure increased very smoothly until it reached its maximum.

Torque build-up was delayed beyond t_1. Most pneumatically actuated clutches squeeze their plates together by pushing (using the force of the air) from one side of the plate assembly while the other side remains fixed. Again, as in the electromagnetically actuated clutch, a finite gap must be crossed resulting in a time delay. Pressure must build up to the point where it will move the member, and then the member must actually move across the gap and contact the plate. Completion of this act is indicated by t_2.

Torque build-up for the two pneumatically actuated disc clutches tested can be described by two adjoining straight lines with different slopes. The initial slope was very steep, suddenly changing to a more

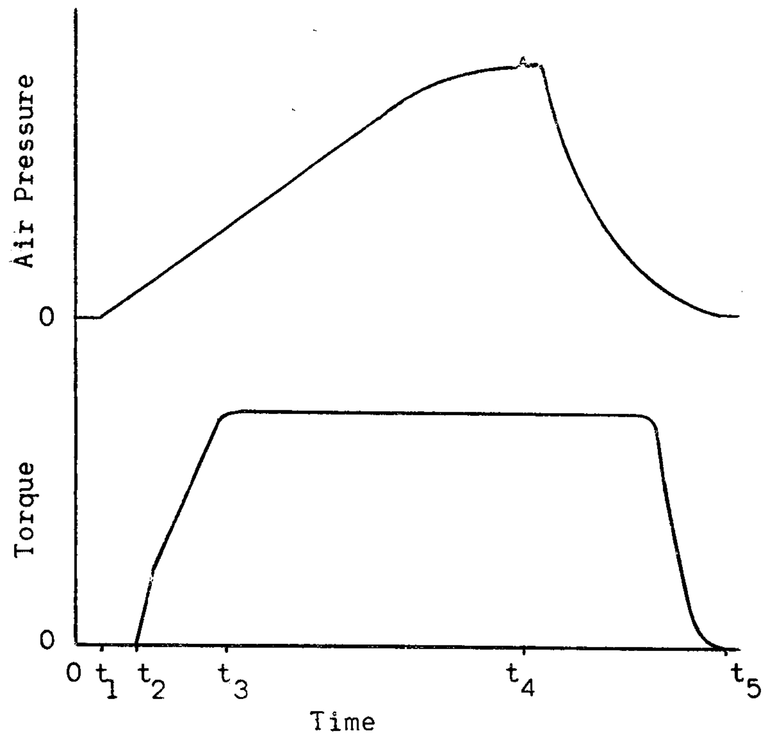

Figure 9. Torque and Pressure History of Pneumatically Actuated Clutches.

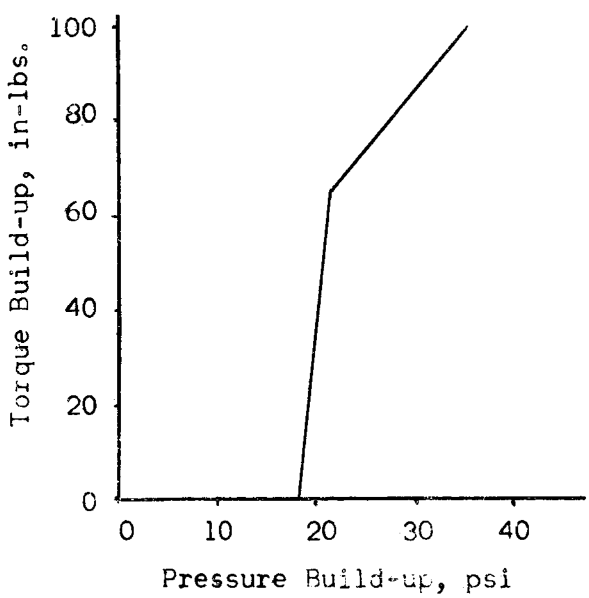

Figure 10. Torque vs. Pressure Build-up of Clutch F.

gradual slope at higher torques. The initial steep slope is thought to be due to the reasonably high pressure initially forcing the plates together. The torque build-up soon overtakes the slower increasing pressure, and the increase in torque per unit time slows to be proportional to the increase in pressure per unit time. The change in slope is noticeable in Figure 10.

At time t_3, torque was sufficient to turn the load at the input speed of the clutch resulting in steady state conditions.

The switch was opened at t_4, picked up electrically and shown on the oscilloscope as a "spike." The valve, however, did not close immediately, for the flux had to deteriorate sufficiently to allow the spring of the solenoid to overpower the high pressure air. As soon as the valve closed, air began exhausting out the exhaust orifice of the valve. Pressure dropped quite low before any change in torque was observed. Both clutches had springs to speed up their release, and as can be seen in the figure, once torque began to drop, its deterioration was very rapid.

Actual data obtained from the tests on the two pneumatically actuated disc clutches are presented in Appendix B, Figure 42 through Figure 44.

It is to be noted that, if desired, smooth and "cushioned" engagements and disengagements can be achieved with pneumatically actuated clutches by using a flow control valve.

For both the pneumatically and the electromagnetically actuated clutches, time response was essentially the same for all speeds. Tests were run with the clutches at rated voltage (pressure) and rated loads, and speeds varying from 200 r.p.m. to 1800 r.p.m. Torque build-up was slightly slower at the lower speeds, but not outside the band of response curves obtained when the clutches were repeatedly cycled at constant speed.

Wrap-Spring Clutches

Some of the characteristics of the wrap-spring clutches are unique, but in general their time history of torque is very similar to an electromagnetically actuated disc clutch. Use of actuating power is one basic difference between wrap-spring and disc clutches. Disc clutches depend on their actuators to produce the required force between the plates. The torque capacity of the disc clutch is directly related to the strength of its actuator. The strength of the actuator of a spring clutch, as long as it is sufficiently strong to trigger the clutch, does not determine the torque capacity. Spring clutches, once actuated, depend on the force produced by their wrapped spring.

Time history of torque for the wrap-spring clutch depends somewhat on the type of actuation, so a description of the clutches tested is given here.

One of the three wrap-spring clutches was actuated electromagnetically. This clutch was designed with the spring attached only to the input. When voltage was applied to the coil, the electromagnetic field that was developed caused a sleeve to force the spring against the output, wrapping the spring.

The other two spring clutches tested operated in an entirely different manner. Although actuation could be considered electromagnetic, the actuator was not an integral part of the clutch. It was instead an external solenoid (frequently not even furnished by the clutch manufacturer) that caught or released a detent protruding from a sleeve around the clutch. The spring was wound such that it was in interference fit with both input and output. One end of the spring was hooked into the sleeve mentioned above. With the sleeve released, the clutch was energized. When the sleeve was arrested by the actuator, the clutch was de-energized and no torque transmitted. For testing, a solenoid was positioned on each clutch so that the clutch was de-energized when the solenoid was de-energized.

Figure 11 shows the general time history of torque and current for the latter type of actuation using a d.c. solenoid. The variation observed when the first type of actuation was used is described in this discussion. Actual data obtained is presented in Appendix B.

Current in the solenoid began to increase immediately upon closing the switch at $t = 0$. At $t = t_1$ the solenoid had completely disengaged from the detent. Torque did not begin to build up immediately, however, for the spring had to wrap. As the spring wrapped, torque began to increase, slowly at first and then very rapidly until the friction load was turned at input speed, $t = t_3$. The torque build-up was essentially the same for the clutch with the actuator an integral part of its construction.

At $t = t_3$ the control switch was opened. Current decayed almost immediately. Depending on the position of the detent there was a time delay from t_3 to t_4 before the solenoid contacted the detent and forced the spring to unwrap. Once torque began to decrease the decay was very rapid.

For the clutch which contained the actuator as an integral part,

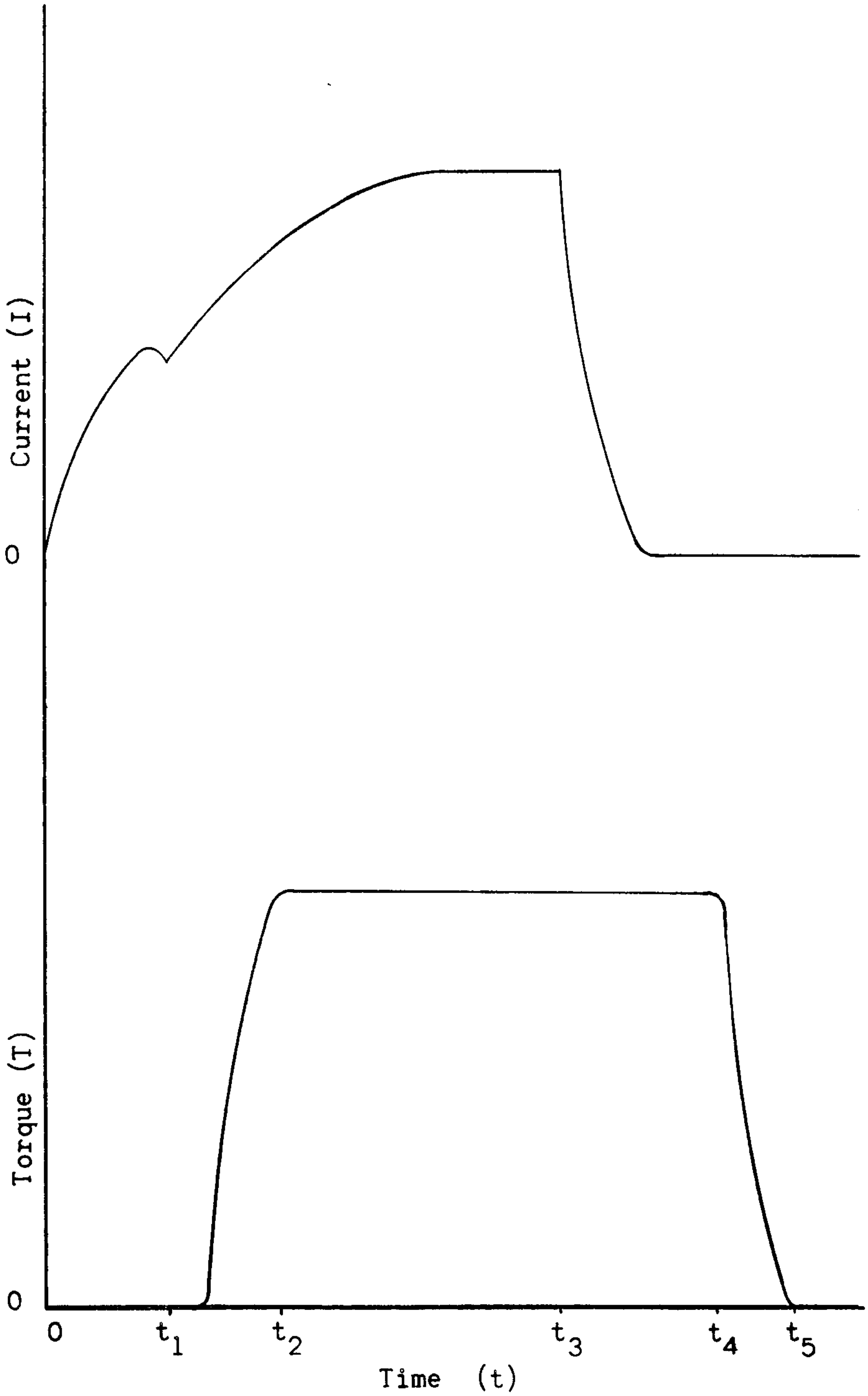

Figure 11. Torque and Current Histories of Wrap-Spring Clutch.

the time delay $t_3 < t < t_4$ would probably have been only long enough for the flux to decay and release the spring. However, actual data for this decay was not obtained because the clutch tested would not disengage. More will be said about this.

The fast response time of the wrap-spring clutch was very impressive. Torque build-up was very fast compared to that of disc clutches, but serious problems were confronted that must be eliminated before the wrap-spring clutch could be used for space purposes.

All three clutches had high drag torques when de-energized resulting in the clutch heating up. At higher speeds this was very significant.

As mentioned above, the clutch whose electromagnetic actuation was part of its construction would not disengage. Consultation with the manufacturer about this problem revealed that the end of the spring had to be started in the proper direction before it would begin to unwrap. This starting action for spring unwrap has been provided in an actuator designed by the manufacturer of the clutch tested.

For the clutches with detents, the time delay between the signal to de-energize and the actual disengagement is not desirable in control systems, especially when the delay is variable, depending on the position of the clutch when it receives the signal. By making clutches available with more than one detent, some manufacturers have partially eliminated this problem. Possibly, a friction surface rather than a detent on the sleeve of the clutch would be the answer. Without any doubt, for applications where precision disengagement is required, the clutches with detents have to be modified.

A general limitation noted concerning spring clutches is that maximum operating speeds of the industrial models are lower than equivalent clutches of other types.

Forcing Circuits

Tests were made to check the influence of forcing circuits. One circuit tested is shown in Figure 13. Specifically, this circuit was used on Clutch E, and components used were a 0.20 microfarad capacitor and a 156 ohm resistor. It was found that the time delay between $t = 0$ and $t = t_1$ shown in Figure 7 was decreased by 30 per cent by using this forcing circuit.

The actual time histories of current are shown in Figure 14. The curve taking the longest time is for Clutch E actuated with only an external resistance (see Figure 12) of 23.9 ohms, used to record the build-up. Steady state voltage across the clutch was 24 volts d.c., the rated voltage for the clutch. The faster curve is for the same clutch, but with the forcing circuit described above. Steady state voltage across the clutch was again 24 volts. To accomplish this, the supply voltage had to be 46.8 volts d.c.

Wear and High Temperature Effects

Since all clutches were tested new, as received from the manufacturers, it was desirous to determine if characteristics changed as the clutch was used.

Clutch C was chosen for the test because of the manufacturer's claim of extremely good wear properties of the friction surface which is a ceramic material. With this clutch set up in the apparatus, the output end of the clutch was fixed and pressure to the hydraulic motor was

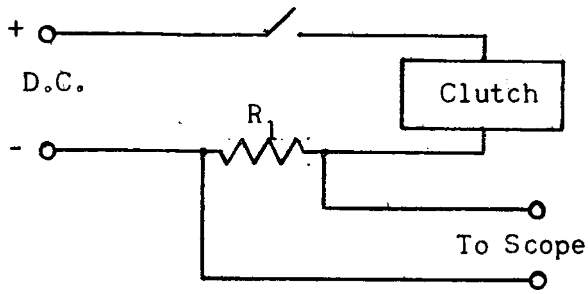

Figure 12. Actuation Without Forcing Circuit.

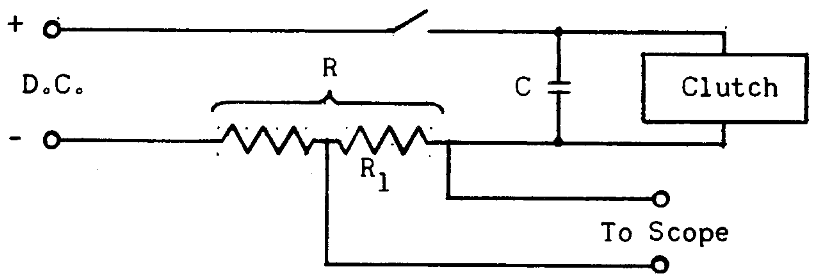

Figure 13. Actuation With Forcing Circuit.

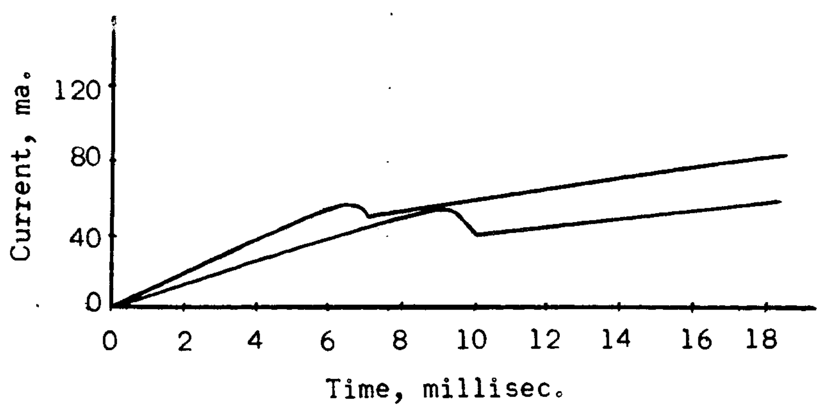

Figure 14. Current Build-ups With and Without Forcing Circuit.

reduced to 250 psi. The clutch was actuated with a square wave generator and a solid state amplifier such that rated voltage was applied to the clutch for 200 milliseconds once every second.

The clutch actually worked as a brake, stopping the input inertia (including the flywheel and the inertia of the hydraulic motor) rotating at about 60 r.p.m. plus the torque generated by the motor. During the off time of the clutch, input speed again became 60 r.p.m.

During each cycle the clutch slipped for about 150 milliseconds. Peak torque reached 125 inch-pounds, but average torque was 75 inch-pounds, only slightly above that rated for the clutch. For fourteen hours the clutch was cycled once each second, resulting in over 50,000 cycles.

Only 0.0005 inch wore from the friction plate of Clutch C after it had cycled over 50,000 times. Very little wear product resulted. The clutch was tested for torque capabilities, and no change was found from those established when the clutch was new. This indicated that 50,000 cycles was only a fraction of the total clutch life.

Clutch C was then subjected to a heating test. An oven was positioned on the test stand such that it surrounded the clutch, but allowed it to turn. The clutch was heated until it reached a steady state temperature of 165 F. The insulation and the coil were rated by the manufacturer for below 175 F. Torque tests were then performed on the clutch.

Rated torque was easily obtained at the high temperature, and time responses seemed unaffected. Maximum torque for the clutch (usually somewhat higher than rated torque) had decreased somewhat at the high temperature. Slightly less current was drawn, too, as might be expected due to the increase in resistance.

Sources of Error

To record torque, the shaft of the torque transducer had to be twisted. In twisting the shaft, all inertia between the clutch and the transducer, including that of the output of the clutch itself, had to be accelerated. The torque required to accelerate this inertia would not be recorded on the oscilloscope, thus torques recorded were accordingly in error. If the inertia between clutch and torque transducer were large and acceleration very fast, significant error could result during the periods of transient torque. An error analysis presented in Appendix C shows that for the system used, the error that might have occurred was quite insignificant.

The Sanborn preamplifiers had rated rise times of one millisecond. Torque history and, for pneumatically actuated clutches, air pressure history were recorded through these preamplifiers.

Although a microswitch was used to actuate the clutches, at the time of actuation intermittent contact was briefly recorded on the oscilloscope. This indicated some bouncing of the contacts within the switch.

CHAPTER III

CLUTCH HEATING

Heat Transfer to a Clutch

The number of cycles per unit time that a clutch can be actuated depends primarily on two considerations. First, the time response of the clutch limits the number of cycles. Physically the clutch can only cycle so many times in a given length of time for a particular actuating force. Secondly, the amount of heat generated internally may be more than can be dissipated during the considered unit of time. This second limitation is often the more serious of the two.

Heat dissipated within a clutch that is cycling is due primarily to slipping, Q_s. If the clutch is actuated by a solenoid, the energy dissipated from the coil also results in heat, Q_e. The total heat generated within the clutch, Q_g, is the sum of Q_s and Q_e.

Heat Due to Slipping

The heat generated due to slipping can be derived as follows: Work for a rotating body can be represented by the equation

$$\delta W = T\,d\theta = T\,\frac{d\theta}{dt}\,dt = T\,\dot{\theta}\,dt \tag{3.1}$$

where

W = work

T = torque

θ = angular displacement

For mechanical-contact clutches, it was previously stated that torque increased, after an initial time delay, essentially linearly with respect to time as shown in Figure 15. Consider time beginning when torque begins to increase, and at $t = t_1$ the torque build-up is complete and is equal to the rated torque, T_r.

Now

$$T = kt \tag{3.2}$$

where

$$k = T_r/t_1 \tag{3.3}$$

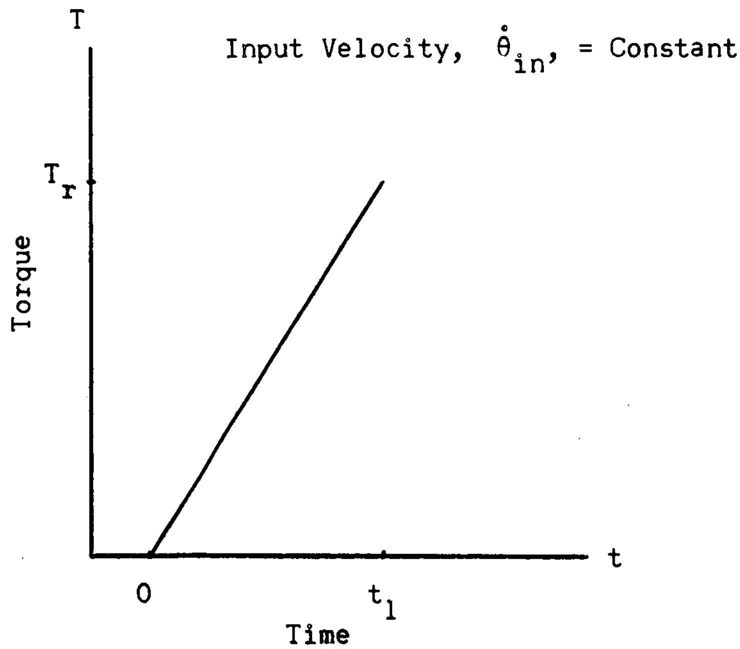

Figure 15. Torque Build-up of Mechanical-Contact Clutches.

Work, therefore, from $t = 0$ to $t = t_1$ is

$$W_{o-t_1} = \int_0^{t_1} T\,\dot{\theta}\,dt = k \int_0^{t_1} \dot{\theta}\,t\,dt \tag{3.4}$$

Consider, now, only the input of the clutch and the work that is developed there. Input velocity was constant for Figure 15, thus work into the clutch is given by:

$$W_{in} = (k\dot{\theta}_{in} t_1^2)/2 \tag{3.5}$$

For an inertia load, torque is given by the relationship,

$$T = I\,\ddot{\theta} \tag{3.6}$$

where

I = inertia of the load

$\ddot{\theta}$ = acceleration of the load

Remember that it was found that the torque build-up for an inertia load was the same as for a friction load. Combining equations (3.2) and (3.6),

$$\ddot{\theta} = kt/I \tag{3.7}$$

Integrating,

$$\dot{\theta} = \int_0^t (kt/I)dt = kt^2/2I \tag{3.8}$$

Note that this is $\dot{\theta}$ of the output shaft.

Placing equation (3.8) into equation (3.4) and integrating, the work out of the clutch between $t = 0^-$ and $t = t_1$ is found to be

$$W_{out} = k^2 t_1^{\,4}/8I \tag{3.9}$$

At t_1, the torque build-up has been completed. Mechanical-contact clutches, if run within their capacities, do not slip once torque has completed its build-up. Thus at t_1 the output velocity equals the input velocity of the clutch. Torque, as indicated in Figure 15, is T_r. As can be seen from equation (3.8), to have the torque build-up completed at $t = t_1$, the inertia of the load must be

$$I = \frac{k t_1^{\,2}}{2\mathring{\theta}_1} \tag{3.10}$$

where

$$\mathring{\theta}_1 = \mathring{\theta}_{in} = \mathring{\theta}_{out} \text{ at } t_1$$

Substituting equation (3.10) into equation (3.9), it is found that the work out of the clutch is

$$W_{out} = \frac{k\,\mathring{\theta}_{in}\,t_1^{\,2}}{4} \tag{3.11}$$

The energy lost to heat is the difference between the work put into the clutch and the work taken out during a given time. Since the clutch does not slip after time t_1, heat generated due to slipping only occurs during the torque build-up. Subtracting equation (3.11) from equation (3.5), it is found that the heat generated due to slipping during t_1 units of time is

$$Q_s = \frac{k\dot{\theta}_{in} t_1^2}{4} \tag{3.12}$$

The spring clutches tested all had drag torques, i.e., torque was transmitted when the clutch was de-energized. These drag torques were not sufficient to turn the load, so all the energy went to heat. For constant input velocity and constant drag torque, during the time interval $t = t_1$ to $t = t_2$ the heat generated in the clutch would be

$$Q = \int_{t_1}^{t_2} T\,\dot{\theta}_{in}\,dt = T\,\dot{\theta}_{in}(t_2 - t_1) \tag{3.13}$$

where T is the drag torque.

<u>Heat Due to Electrical Dissipation</u>

For an R-L circuit like that shown in Figure 16, current builds up according to the well known equation

$$I = \frac{E}{R}\,(1 - e^{-(R/L)t}) \tag{3.14}$$

as is illustrated in actual test data. As can be also seen in the data, once the switch is opened, current dissipates almost immediately (see Appendix B).

In order to obtain a general equation for the heat generated per cycle in the coil, some assumptions must be made. Assume that no heat is generated after the switch is opened. Assuming the applied voltage is constant, and neglecting the energy (work) used to move the plates to together, the heat generated in the coil per cycle of the clutch is proportional to the area under the curve of the time history of current. Specifically,

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Page 33: Equation (3.15) should be

$$Q_e = \frac{E^2}{R}\left[t_1 - \frac{2L}{R}\left(1 - e^{-\frac{R}{L}t_1}\right) + \frac{L}{2R}\left(1 - e^{-\frac{2R}{L}t_1}\right)\right] \qquad (3.15)$$

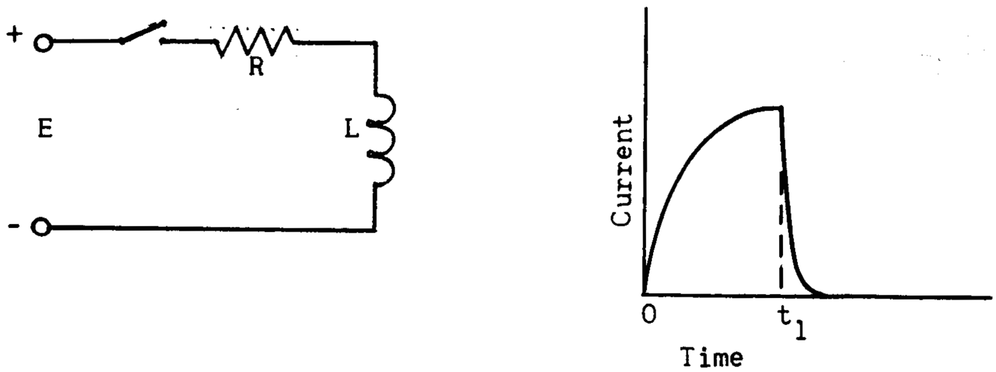


Figure 16. An R-L Circuit and Its Current History.

Heat Dissipation From a Clutch

Heat dissipated from a clutch varies with the size of the clutch and the manner in which it is mounted. A clutch operating on earth in an atmosphere dissipates much of its heat by convection. In space, however, the clutch may be operating without an atmosphere or without a gravitational force, both being required for free convection. Radiation and conduction are the other means by which heat transfer can be accomplished.

Some clutches are mounted like couplings with no fixed base or support. For this case, radiation is the prime factor in transmitting heat. C. Depken (24) has presented a detailed discussion of this, and only a brief summary will be presented here. When the clutch is mounted on a base, heat transfer by conduction can be added to that transferred by radiation. In a vacuum, however, an important consideration in determining heat transfer by conduction is thermal contact conductance between mating surfaces, i.e., the ratio of heat transferred across a joint per unit time, per unit area of the material surfaces, per unit temperature drop across the surfaces. A detailed discussion of this topic will be presented.

Heat Transfer by Radiation

For Depken's analysis, the clutch was assumed to be contained without connections within a surrounding container whose temperature was a constant 70 F. Assuming that irradiation for both surfaces was uniform, for a gray body the heat transfer by radiation from the clutch per unit area of the clutch is given by the equation

$$\frac{Q}{A_1} = \frac{\sigma(\bar{T}_1^{\,4} - \bar{T}_2^{\,4})}{\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)} \qquad (3.16)$$

where subscript "1" refers to the clutch and subscript "2" refers to the surrounding surface. Assuming $\varepsilon_1 = \varepsilon_2$ and that $\bar{T}_1 = 654$ R based on rated temperatures for the insulation of industrial clutches, Depken presented curves showing heat transfer per unit area for ε values from 0.1 to 1.0 and for area ratios from 0.0 to 1.0.

Heat Transfer by Conduction

As shown in any elementary heat transfer book (25), the heat transfer per unit time by conduction, Q_c, through a composite structure such as shown in Figure 17 is given by

$$Q_c = \frac{T_i^o - T_o^o}{\sum_{n=1}^{n=j} R_n} \tag{3.17}$$

Consider a rectangular base on the clutch whose upper surface is the temperature of the clutch, T_c^o (see Figure 17). The base is in contact with the surface of a heat sink at temperature T_s^o. The problem is to determine the heat transfer from the clutch to the sink.

If contact conductance could be neglected, the problem would be simplified greatly, but unfortunately even in an atmosphere and especially in a vacuum contact conductance, h_c, must be considered.

Because surface finishes of metals are usually irregular, physical contact is not made at all points when two surfaces are held together. In fact the actual contact area is usually only a small fraction of the apparent contact area. Not only is the effective contact area a function of the surface finishes, but also a function of the material flatness, the contact material itself, and the contact pressure.

Thermal conductance of a contact or joint is primarily a surface effect depending on the above mentioned conditions of the surfaces, the properties of the materials making the contact, and the interstitial

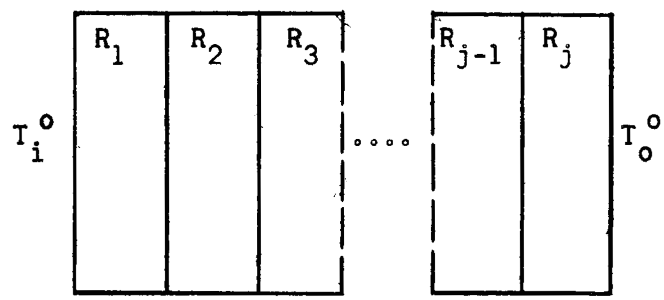


Figure 17. Composite Structure.

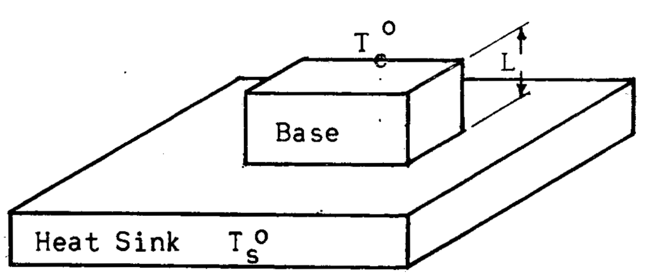


Figure 18. Base On Which Clutch Might be Mounted.

fluid. The modes of heat transfer across the contact are 1) thermal radiation, 2) conduction through the contacting high points of the surfaces, and 3) gaseous, molecular, or other conduction through the interstitial fluid. In vacuum only the first two modes are effective.

Theoretical analysis of this problem is possible and has been done, but many of the terms in derived equations are either difficult to evaluate or are not known for the practical problem. Data, then, must be obtained by experimental means. Fundamental analysis had been made earlier for thermal contact conductance of metals, but until 1962 very little data had been obtained. The data that had been obtained was primarily for metals in air and sometimes other gases. Some of these are referenced in the Bibliography. In 1962 E. Fried and F. A. Costello (26) analyzed the thermal contact conductance problem as applied to unpressurized satellites and components. Data was presented on the effects of surface finish and flatness and on the contact conductance between aluminum-aluminum and magnesium-magnesium plates at pressures of 10^{-4} to 10^{-6} mm Hg absolute.

In their report Fried and Costello state that it was observed that at zero contact pressure the thermal conductance was approximately the same as when there was a finite gap of several millimeters between the surfaces. This indicates that only radiation and free molecular conduction are important heat transfer paths for this condition.

In vacuum, and for contact pressures up to 35 psi, it was found that contact conductances were of the order of 20 to 125 Btu/hour-square foot-degree F. Contact conductance was considerably improved when relatively soft shim materials (lead and aluminum) were introduced between the surfaces. Other methods suggested to improve contact conductance were 1) increasing contact pressure, 2) reduced surface roughness, and 3) improved flatness of surfaces.

In 1963, H. Fench and W. M. Rohsenow (27) analyzed heat conduction through surfaces in contact, deriving an equation for contact conductance. In 1964, H. Fench and J. Henry published a paper on "The Use of Analog Computers for Determining Surface Parameters Required for Prediction of Thermal Contact Conductance" (28). Only a brief section was devoted to vacuum operation, however.

Work done by W. R. Stubstad (29) indicated that in vacuum a combination of rubber and oil (RTV-S-5313 and 5314 with 25 per cent by weight Dow Corning 200 silicone oil) very significantly increased contact conductance. In Stubstad's tests, the surfaces were first exposed to vacuum before being placed in contact. He indicated that lower contact conductances were obtained in vacuum when surfaces were initially placed in contact before evacuating the chamber. Data on this point were "inconsistent . . . and therefore . . . not presented" in the publication (30). A graph obtained by Stubstad is shown in Figure 19 indicating how contact conductance varied with the environmental pressure when mechanical pressure between the plates was a constant 2 psi.

Walter Aron and Gerald Colombo worked on the problem of heat transfer across bolted joints in a vacuum (31). In their report they approximated data obtained by E. Fried and others by a math model relating thermal contact conductance for bolted joints in a vacuum environment to pressure distribution and material softness:

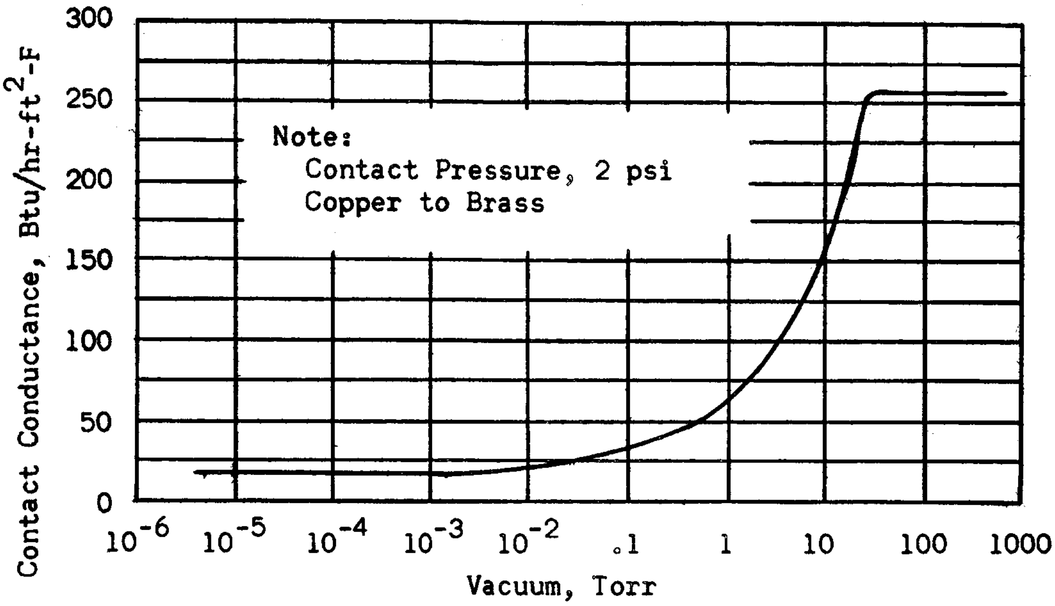


Figure 19. Variation of Contact Conductance with Environmental Pressure.

Table 1. Contact Conductance of Molybdenum-Aluminum Joint

Contact Pressure psi	h_c Vacuum, 10^{-5} mm Hg Btu/hr ft^2 F	h_c Air, 14.7 psi Btu/hr ft^2 F
40	135	167
60	170	195
80	150	178
140	330	355
550	1120	1120
725	1520	1520
1125	2600	2600

$$h_c = 10^4(P/Y_o)^{2/3} \quad \text{for} \quad (P/Y_o) < 0.5 \tag{3.18}$$

where

P = interface contact pressure, psi

Y_o = initial yield stress at elastic limit of plate material, psi

They also investigated the pressure distribution of a bolted joint and found the pressure dropped to zero at a radius approximately equal to the loading radius plus 1.5 to 2.0 times the thickness of the plates.

Petri (32) found that as contact pressure increased, the difference in thermal conductivity between air and vacuum cases became small. Tabulated in Table 1 is data he obtained by testing a molybdenum-aluminum lathe-machined joint in air and in a vacuum of 10^{-5} mm of mercury. The decrease in conductance at 80 psi was caused by his testing apparatus breaking, pitting the surface of the aluminum.

An example will illustrate the use of equations and data and the importance of including contact conductance in conduction calculations.

Assume the base of the clutch in Figure 18 has a height of one inch and is made of aluminum. Assume T_c^o is 200 F and T_s^o is 70 F.

From elementary heat transfer,

$$R_k = L/(AK) \tag{3.19}$$

where in this example L is the height, A is the area of the section through which heat flows by conduction (perpendicular to the direction of heat), and K is the thermal conductivity of the material.

Another equation that will be used is

$$R_h = 1/(h_c A) \tag{3.20}$$

where A is the area of the section through which heat flows and h_c is as defined before, contact conductance.

For this example, then,

$$Q_c/A = \frac{T_c^o - T_s^o}{(1/h_c) + (L/K)} \tag{3.21}$$

K for aluminum is approximately 120 Btu/hr ft F. Assuming the heat sink is molybdenum and the contact pressure is about 50 psi, h_c would be approximately 150 Btu/hr ft^2 F. Using these values and 1/12 foot for the height of the base, the heat transfer per unit area is about 17,700 Btu per hour per square foot. If the effect of contact conductance had not been included, the result would have been 188,000 Btu/hr ft^2. Any other connections, such as between the clutch and the base, would further reduce the heat transfer.

CHAPTER IV

CLUTCH CHARACTERISTICS

Estimated Limit of Torque Per Unit Weight and Per Unit Volume

For comparison purposes, Table 2 is presented giving for each clutch tested the manufacturer's rated torque and the maximum speed at which the clutch should be run. Also tabulated are the rated torque per unit volume and rated torque per unit weight of each clutch. It is important to remember that these clutches were manufactured for industrial purposes and not specifically for use in space. If one of these clutches were to be used in space, it would probably be integrated into a "package," and would not have the weight and volume an industrial clutch has. However, on a comparison basis, Table 1 has something to offer.

Honors for the most torque per unit volume and per unit weight go unquestionably to the wrap-spring clutch. Although at low torques electromagnetically actuated disc clutches have low torque per unit weight and per unit volume, as torque increases above 50 inch-pounds these values become quite good. From data obtained from a manufacturer's catalog (33), an electromagnetically actuated clutch with rated starting torque of 365 inch-pounds has a rated torque per unit volume of 10.2 inch-pounds per cubic inch and a rated torque per unit mass of 91.5 inch-pounds per pound, comparing favorably with the wrap-spring clutch.

Pneumatically actuated disc clutches are usually larger and heavier

Table 2. Rated Characteristics of Clutches Tested

Clutch	Actuation	Type	Rated Working Torque (in-lb)	Max Rated speed (rpm)	Rated Torque per unit volume (lb/in^2)	Rated Torque per unit weight (in)
A	Electromagnetic	Multiple Disc (4)	100	6000	9.61	52.4
B	Electromagnetic	Disc	100*	5000	8.1	45.4
C	Electromagnetic	Disc	70	Unspecified	8.14	40.0
D	Electromagnetic	Disc	16	Unspecified	2.56	16.0
E	Electromagnetic	Disc	1	Engage 500 Run 1000	2.08	1.3
F	Pneumatic	Multiple Disc (2)	210 @ 60 psi	Unspecified	3.12	32.3
G	Pneumatic	Disc	100 @ 60 psi	1800	1.732	16.7
H	Electrical (External Solenoid)	Wrap-Spring	250	500	70.7**	250.0**
I	Electromagnetic	Wrap-Spring	240	1800	14.7	101.6
J	Electrical (External Solenoid)	Wrap-Spring	200	1850	60.4**	230.0**

*Rated Static Torque

**Does not include solenoid

than other disc clutches or spring clutches with similar torque capacities. This is because of their particular applications in industry, many times operating in places where working conditions are much less than ideal. The principle of pneumatic actuation does not require large mass or volume, so for space purposes such a clutch could be designed with less weight and volume than an industrial model. From Table 1, however, it is obvious that pneumatically actuated clutches currently manufactured have poor torque-to-weight and torque-to-volume ratios compared to other clutches.

It is estimated that if an electromagnetically actuated disc or wrap-spring clutch were to be used in space, 10 per cent of the weight and volume of the presently manufactured models could be eliminated. About 25 per cent of the weight and volume of pneumatically actuated disc clutches could be eliminated. Very little, if any, weight or volume could be eliminated from the presently manufactured wrap-spring clutches with detents. Applying this estimation, Figures 20 and 21 show weight and volume for various rated torques based on the clutches tested.

Even with the elimination of 25 per cent of its weight and volume, pneumatically actuated disc clutches still have low torque-to-weight and torque-to-volume ratios.

Response Times

Table 3 presents times required to obtain certain responses of the clutches tested. As can be seen in Figure 22, these response times varied considerably with the clutch. About the only trend that can be

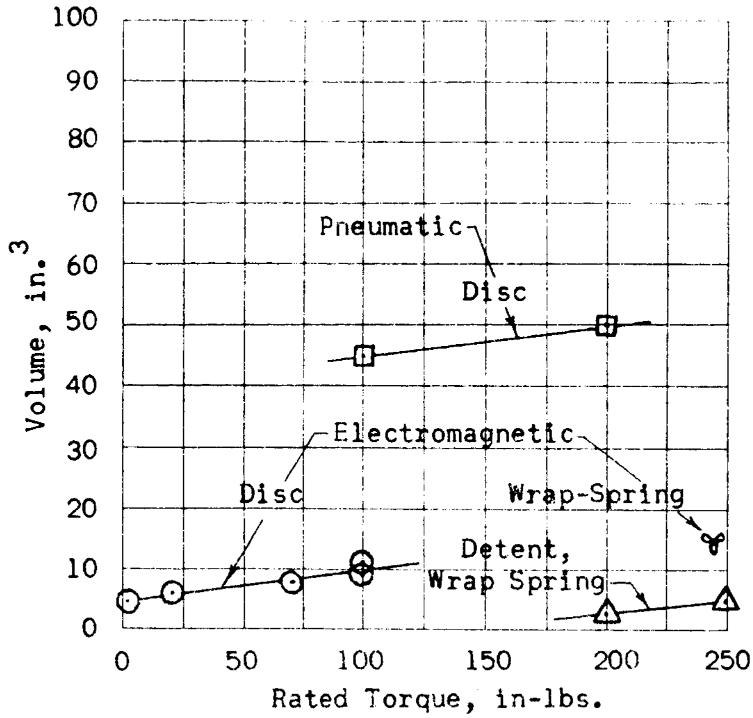


Figure 20. Volume Required for Rated Torque.

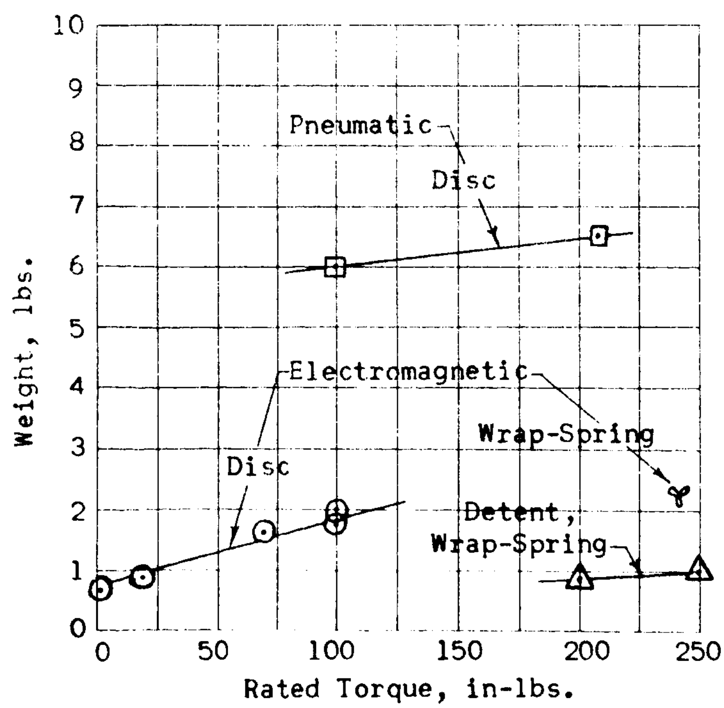


Figure 21. Weight Required for Rated Torque.

pointed out is that the data for the disc clutches indicate a general increase in time response for an increase in torque.

Actuating Power Requirements

Table 4 is presented to illustrate two significant points. It has been mentioned before that the torque capacity of disc clutches depends upon, among other factors, the pressure between the plates. As shown in Table 2, clutches "A" through "E" are electromagnetically actuated disc clutches listed in the order of decreasing rated torque

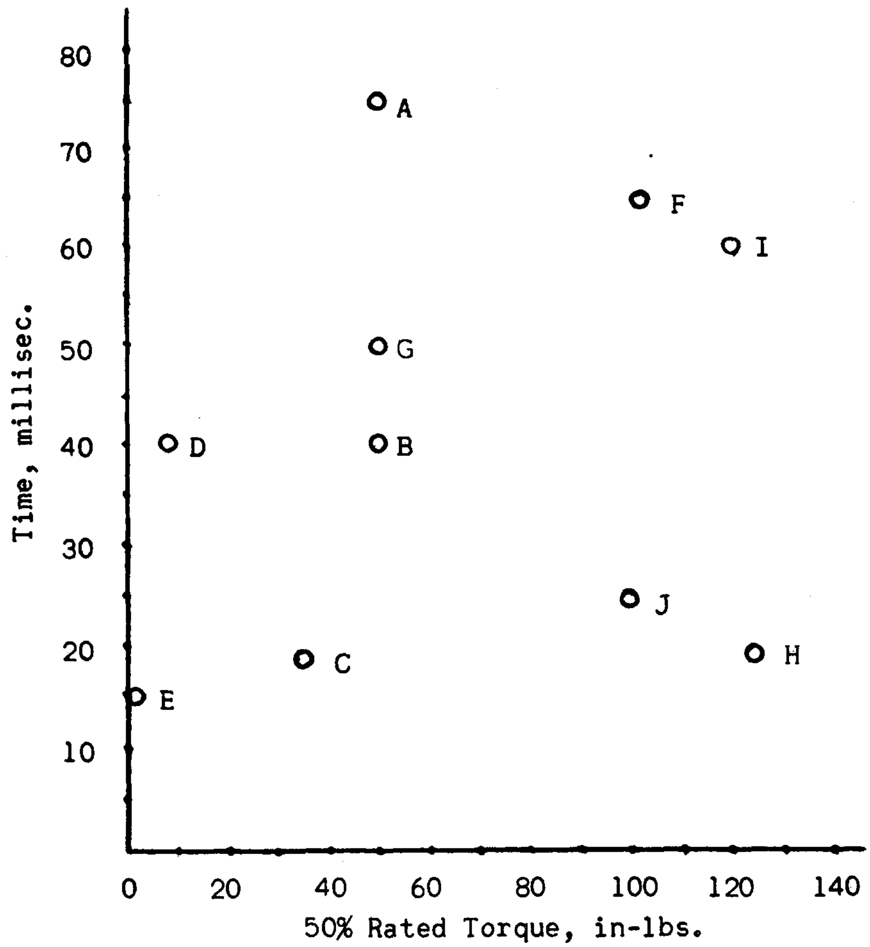


Figure 22. Time Required to Reach 50% Rated Torque for the Various Clutches Tested.

Table 3. Time Responses of Clutches Tested

Clutch	Rated Torque (in-lbs.)	Time Delay from Switch Closed to Beginning of Torque Increase (Milliseconds)	Time from Switch Closed To 50% Rated Torque (Milliseconds)	Time From Switch Closed to 100% Rated Torque (Milliseconds)	Time From Switch Open until Zero Torque (Milliseconds)
A	100	20	75	115	50
B	100	10	40	-	40
C	70	9	18	30	35
D	16	20	45	70	25
E	1	10	15	20	8
F	210 @ 60 psi	38	65	-	35
G	100 @ 60 psi	30	50	95	25
H	250	9	19	-	10
I	240	45	60	-	-
J	200	17	25	-	15

capacities. Not only do these clutches depend upon the electromagnetic field to actuate them, but they also use it to apply the necessary pressure between the plates. Table 4 shows very clearly that for increased torque, the electrical power required to operate the clutch must be increased.

Table 4. Coil Data for Electromagnetic Clutches

Clutch	Rated Torque (in-lbs)	Rated D.C. Voltage (volts)	Coil Resistance (ohms)	Rated Input Power (watts)
A	100	90	670	12.1
B	100	90	865	9.4
C	70	12	17	8.5
D	16	28	145	5.4
E	1	24	175	3.3
I	240	24	320	1.8

That increased operating power is required for increased torque is true for wrap-spring clutches as well, but Table 4 illustrates a significant difference. Manufacturers of wrap-spring clutches use the electromagnetic field primarily to actuate the clutches. Once actuated, the clutch is not limited by the power of its electrical coil. To transmit torque, the clutch uses the radial force of the wrapped spring and the coefficient of friction between the spring and the surface on which it is wrapped. The significance of this is that much less power is needed to operate a spring clutch than is required for a disc clutch with equivalent rated torque. Although disc clutches "A" and "B" have rated torques of less than half that of the clutch "I," a wrap-spring clutch, the power rating of "I" is less than one fifth that of "A" or "B."

Wear Products

A problem encountered occasionally with disc clutches is contamination of the plates in the form of wear products. In space applications, wear products could produce multiple problems by contaminating unprotected instruments in the vicinity of the clutch.

Wear products were particularly noticed after only a few cycles of Clutch A. The driving discs of this clutch were made of steel and the driven discs were of bronze. On the other hand, Clutch C, whose friction material was a ceramic, showed very little wear product even after it had been cycled over 50,000 times.

CHAPTER V

CONCLUSIONS

Time history of torque was found to have the same general form for all clutches tested. Variations were primarily due to delays caused by the means of actuation. Wrap-spring clutches had the fastest response of the clutches tested, but were not as reliable in disengaging as the disc clutches. Of the disc clutches, those electromagnetically actuated had a faster response than those pneumatically actuated. Springs were found to be very effective in decreasing the disengagement time of disc clutches. Controlled torque history can be obtained by using a flow control valve in conjunction with a pneumatically actuated disc clutch or by varying the applied voltage on an electromagnetically actuated disc clutch.

The use of forcing circuits to cause overexcitation of electromagnetically actuated clutches very effectively reduced response times.

Wear of friction clutches sometimes results in products that can cause contamination of plates and a decrease in the torque capacity of the clutch. There are friction materials available, though, that wear very little and whose products of wear are kept at a minimum.

Most disc clutches have very little, if any, drag torque. All three wrap-spring clutches tested had significant drag torques -- high enough to cause heating problems at speeds above 500 r.p.m.

Wrap-spring clutches had the highest torque-to-weight and torque-to-volume ratios. Electromagnetically actuated disc clutches had ratios nearly as high as the ratios for wrap-spring clutches, but the ratios for pneumatically actuated disc clutches were low.

Wrap-spring clutches had much lower actuating power requirements than did disc clutches of equivalent torque capacities.

Maximum speeds are more limited for wrap-spring clutches than for disc clutches.

Because manufacturers tend to conservatively rate the torque capacities of their clutches, there is no problem in obtaining the rated torque of the clutch even at high temperatures. Maximum torque for the electromagnetically actuated disc clutch tested decreased, though, when the clutch was operated at high temperatures.

Heat generated within a clutch can be calculated knowing the slipping characteristics of the clutch as a function of time and the values of the components making up the electrical network if they are actuated electrically.

Heat can be transferred from a clutch by radiation, conduction, and convection. Without an atmosphere or a gravitational force, only the first two apply.

Contact conductance should be considered when making conduction calculations, especially if the application is to be in vacuum.

APPENDIX A

INSTRUMENTATION AND EQUIPMENT

A block diagram and a general description of the test apparatus were given in Chapter II. Detailed description of the instrumentation and equipment is given here.

A Vickers AR-10007-Bel hydraulic motor (characteristics similar to the Vickers Model 913 aircraft-type hydraulic pump) was used, controlled by a Moog servo valve, Model 22-135A. A Denison hydraulic pump was used to supply fluid to the valve at a pressure of 1000 psi.

Valves (see Figure 23) were used to completely by-pass the servo valve, if desired. High speeds could be obtained with the by-pass valves and accurately controlled with the servo valve.

A flywheel was used directly after the hydraulic motor to reduce fluctuations in speed due to sudden applications of loads.

The speed of the apparatus was displayed on a Hewlett-Packard 52331 Electronic Counter. This was accomplished by having a gear with 120 gear teeth turn on a shaft connected to the flywheel shaft. As the gear turned, a photosensitive pickup focused on the gear gave a pulse for every gear tooth that passed in front of it. The pickup was Model 836 by Power Instruments Inc. The pulses were summed and displayed in intervals of one second each by the Hewlett-Packard counter. Since the gear had 120 gear teeth and the pulses were counted for one second, the number displayed was numerically equal to twice the speed of the apparatus in r.p.m.

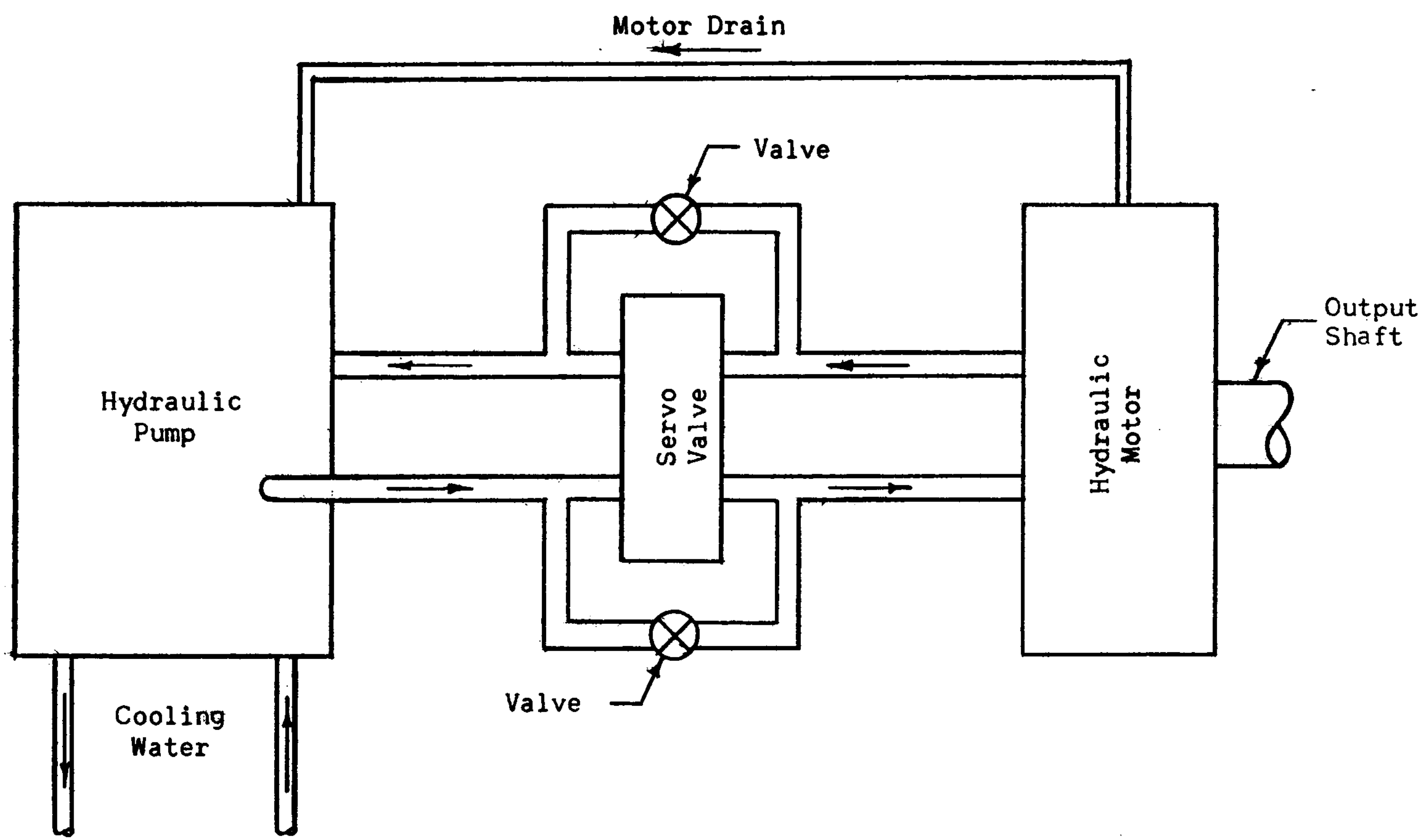


Figure 23. Piping Diagram.

Another gear turned on the same shaft as the gear used to read the speed. This second gear was used to turn a generator which put out a d.c. voltage proportional to speed. The generator was the first element of the feedback circuit. Made by Fairchild Industries (Model 532-A), it generated 0.00357 volts per r.p.m. of the hydraulic motor when 12 volts were applied across the field.

To filter out unwanted a.c. signals, a filter such as shown in Figure 24 was used on the output of the generator. Components were used (see Figure 26) such that τ_1 was 0.000156 and τ_2 was 0.016, making the filter's transfer function:

$$E_o/E_i = \frac{0.000156S + 1}{0.016S + 1}$$

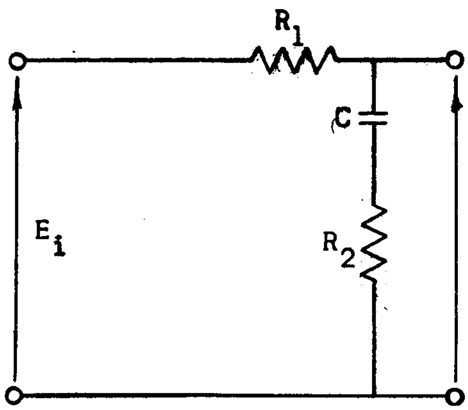
As long as very little current went through the filter, there was little drop in the signal.

The feedback loop was summed with the reference voltage. A circuit as shown in Figure 25 was used to accomplish the summing. With components as shown in Figure 26, the output of the summing circuit was:

$$E_o = (E_s + E_g)/12$$

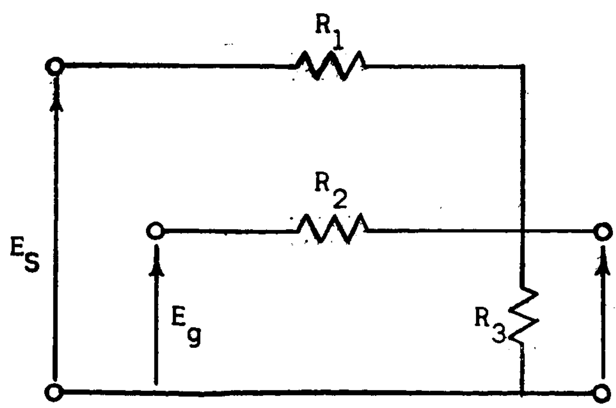
The voltage output of the generator was of opposite sign than that of the supply voltage, resulting in a "negative" feedback control system.

The output of the summing circuit went into a Dymec amplifier, Model 2460A. The input impedance of the amplifier was one megohm at d.c., so very little current was drawn from the supply and the generator.



$$E_o = \frac{\tau_1 S + 1}{\tau_2 S + 1} E_i$$

Filter 24. Filter Circuit.



$$E_o = \frac{R_3(R_2E_S + R_1E_g)}{R_1R_2 + R_1R_3 + R_2R_3}$$

Figure 25. Summing Circuit.

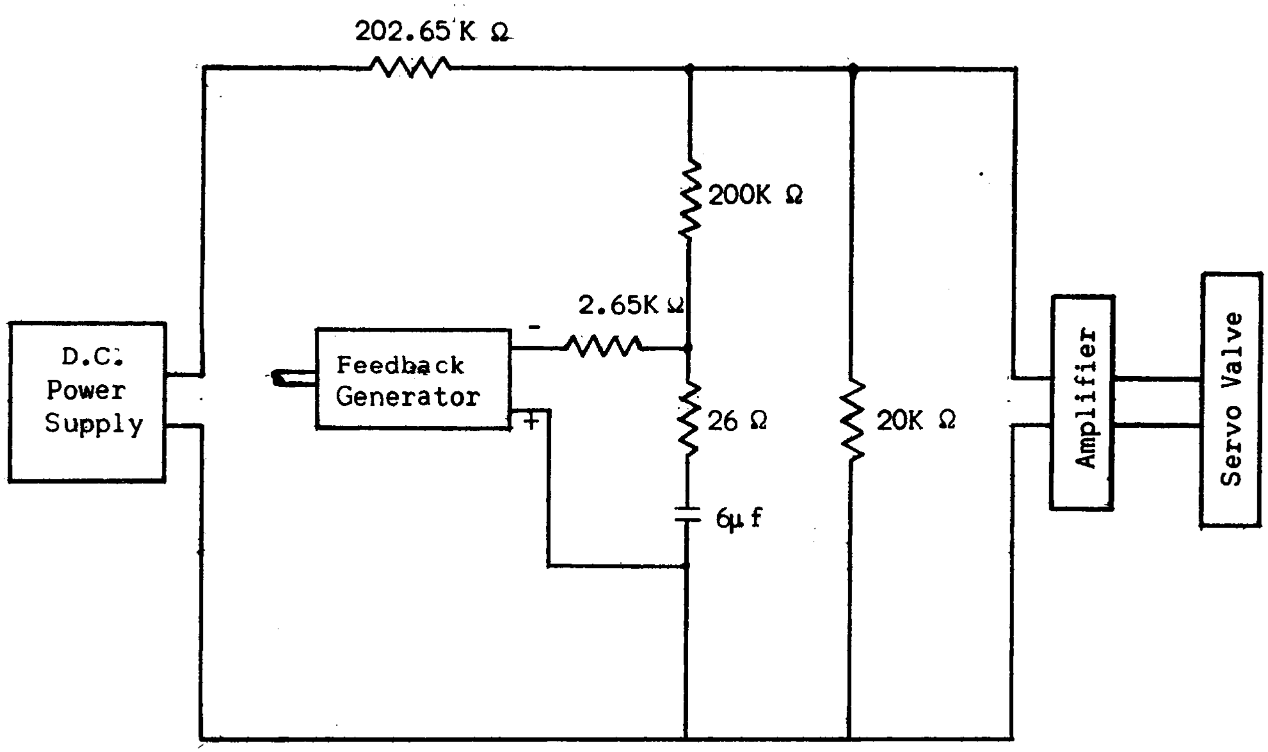


Figure 26. Feedback Control Wiring Diagram.

Output of the amplifier was connected directly to the servo valve to control the motor.

Three d.c. power supplies were used, one of which was made according to the diagram in Figure 27. This supply was used to supply the 12 volt field for the d.c. generator. Another supply, Model 6204A by Harrison Laboratories, was used as the reference signal for the entire system. The third power supply was one half of a Kepco Model 430D, voltage-regulated dual d.c. power supply and was used to supply power to the photosensitive pickup. The other half of the dual power supply was used to furnish power to electrically actuated clutches.

The clutch to be tested was positioned after the feedback loop. Air, a.c. power, and d.c. power as mentioned above were available for actuating clutches.

A Model 1214-200 Lebow torque transducer, with torque capabilities up to 200 inch-pounds, was used for the larger clutches. With capabilities only up to 200 inch-ounces, Model MTE-200, also by Lebow, was used to test the smaller clutches.

A Type 564 Storage Oscilloscope with a Type 3172 Dual-trace Amplifier, both made by Textronix, Inc., was used to record the time history of torque. The scope was triggered by the switch that actuated the clutch being tested, and a Sanborn recorder preamplifier, Model 350-1100B, was used to amplify the transducer's signal to a level suitable for the scope.

Between the preamplifier and the scope, a low-pass filter was used to remove high frequency vibrations generated by the test apparatus during torque build-up (see Figure 28). This filter began attenuating

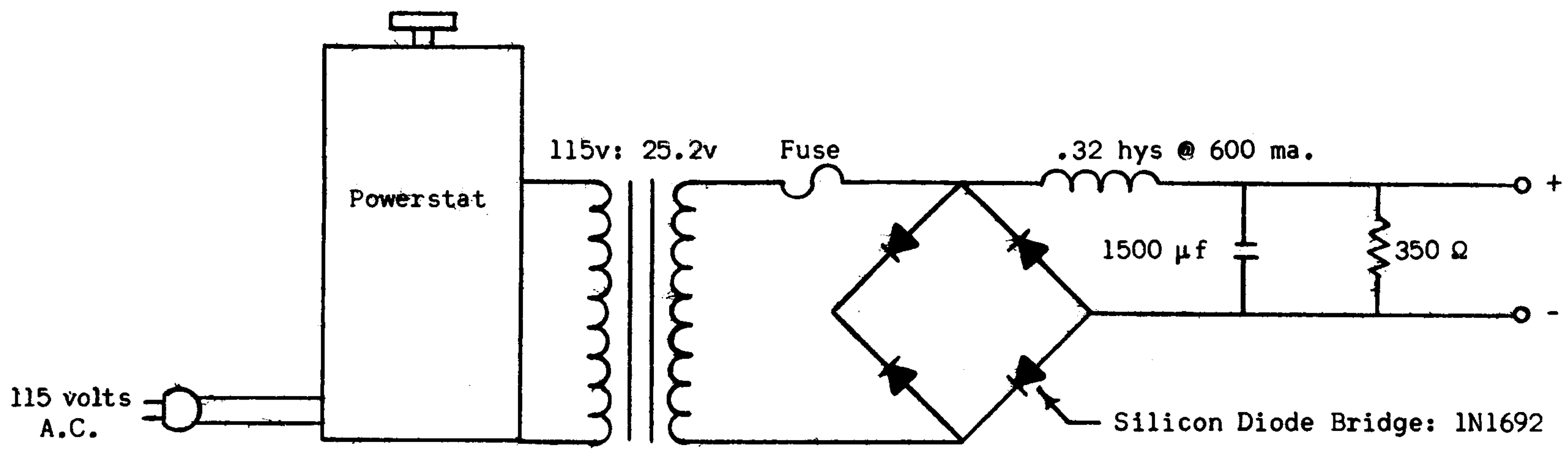


Figure 27. D.C. Power Supply (600 ma.).

signals with frequency of about 50 cps. Before applying the filter, small vibrations having a frequency of 350 to 400 cps were observed (this agrees very well with the error analysis in Appendix C). The filter eliminated these vibrations. To damp out some of the vibrations and to provide a solid test stand, the apparatus from servo valve to fixed end was mounted on a lathe bed.

Although air-operated clutches were actuated by a solenoid valve (ASCO No. 831723 3-way valve), it was the pressure history rather than the current history of the solenoid that was of interest for these clutches. To obtain the pressure history, a Baldwin-Lima-Hamilton Corp. pressure transducer, Type GP-CG, was placed in the air line to the clutch. Another Sanborn Model 350-1100B preamplifier was used to amplify the transducer's signal before it entered the oscilloscope.

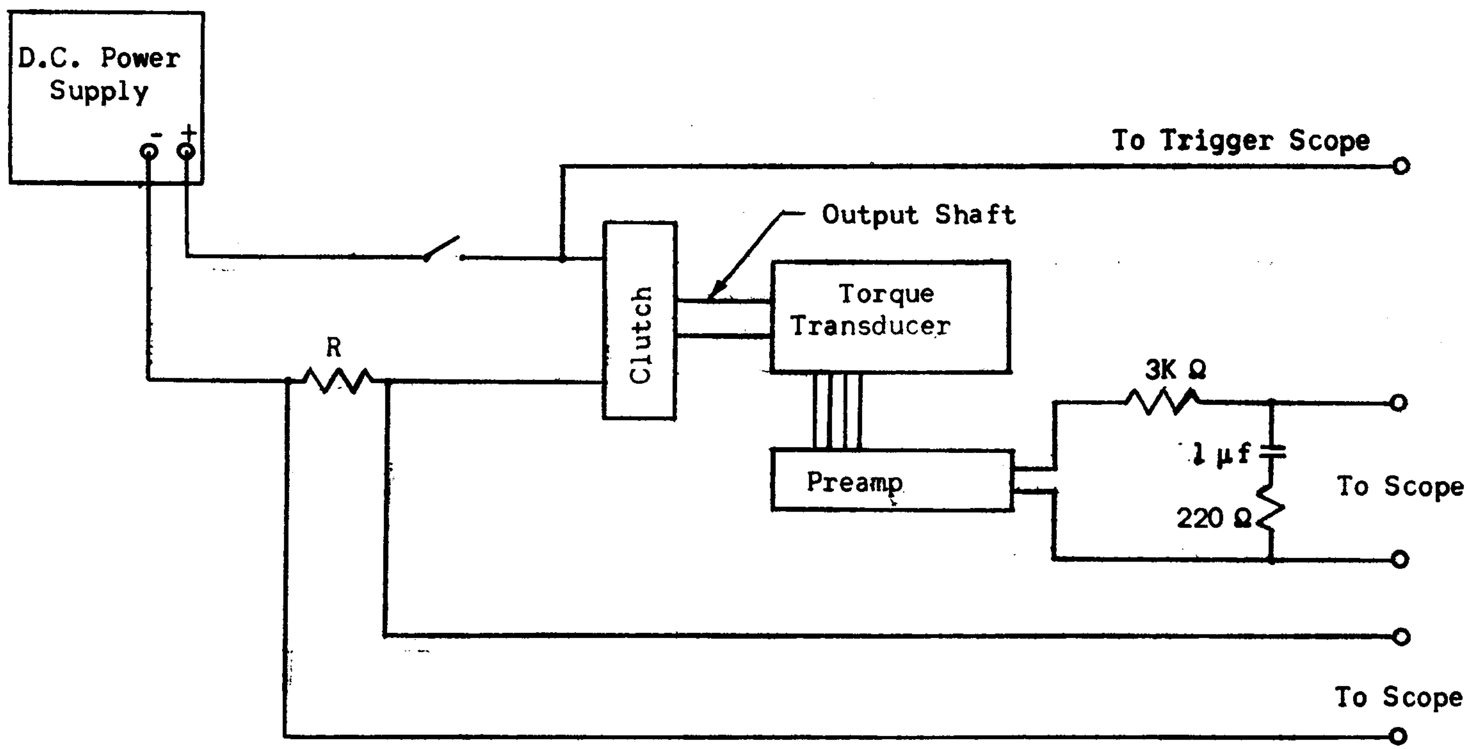


Figure 28. Test Setup for Electrically Actuated Clutches.

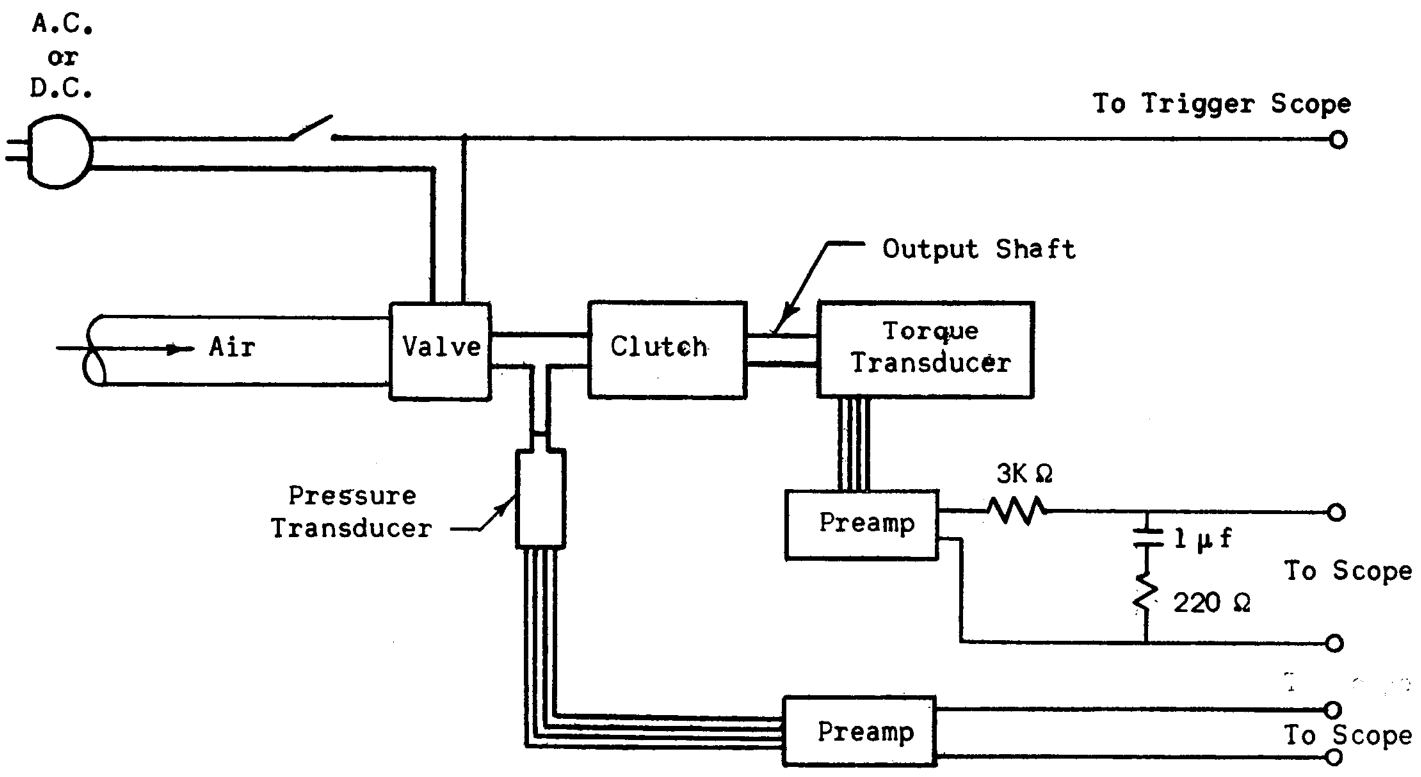


Figure 29. Test Setup for Pneumatically Actuated Clutches.

APPENDIX B

EXPERIMENTAL DATA

On the following pages are representative data obtained in tests. A brief description of each figure is given.

Figure 30 is the time history of torque and current for Clutch A. The input speed of the clutch was 1000 r.p.m.

Figure 31 shows three time histories of Clutch A (made by triple exposure), changing each time the voltage applied to the clutch. For the curve "a," applied voltage to the clutch was 125 volts d.c. Rated voltage, 90 volts d.c., was used to obtain the curve "b." Curve "c" was obtained when the voltage supplied to the clutch was 60 volts. Input speed was held constant at 1000 r.p.m. for all three runs.

The effect of a forcing circuit on Clutch A is shown in Figure 32. The circuit used was of the same type as shown in Figure 13. The values of the components were 0.22 microfarads and 465 ohms. Curve "a" is of the clutch actuated without the forcing circuit. To obtain the curve "b," the forcing circuit was utilized.

Representative data for Clutch B are shown in Figure 33 through Figure 35. Torque and current time histories are shown in Figure 33. For these histories the input speed of the clutch was 1000 r.p.m., and the voltage applied across the clutch was 90 volts d.c. The same input speed and applied voltage were used to obtain Figure 34. For this run, however, the torque at which the slip clutch would slip was set much

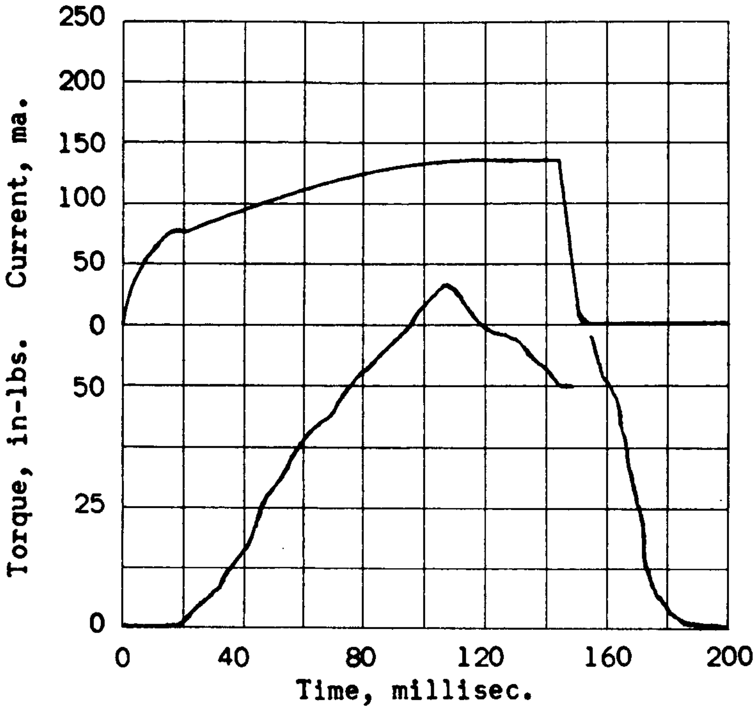


Figure 30. Histories of Clutch A.

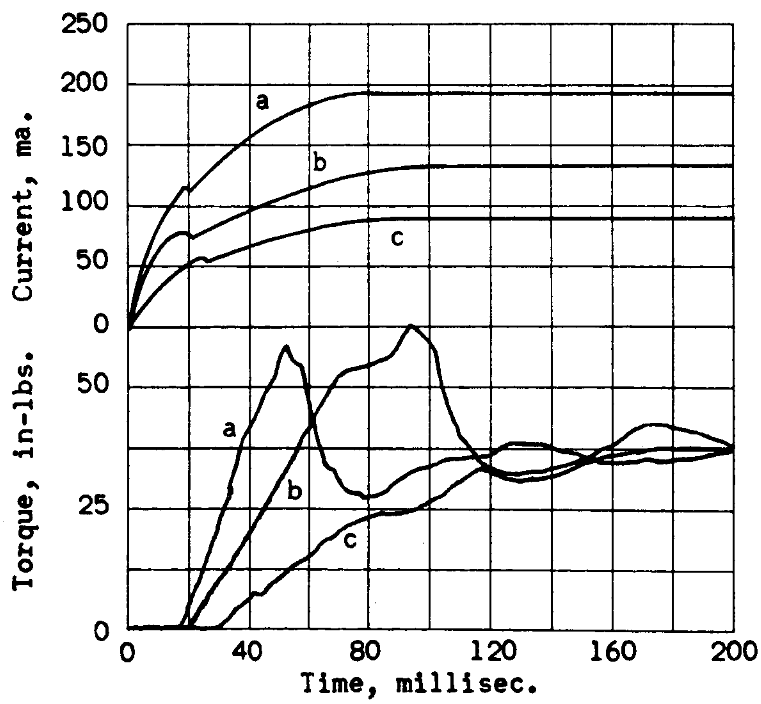


Figure 31. Variation of Histories with Voltage of Clutch A.

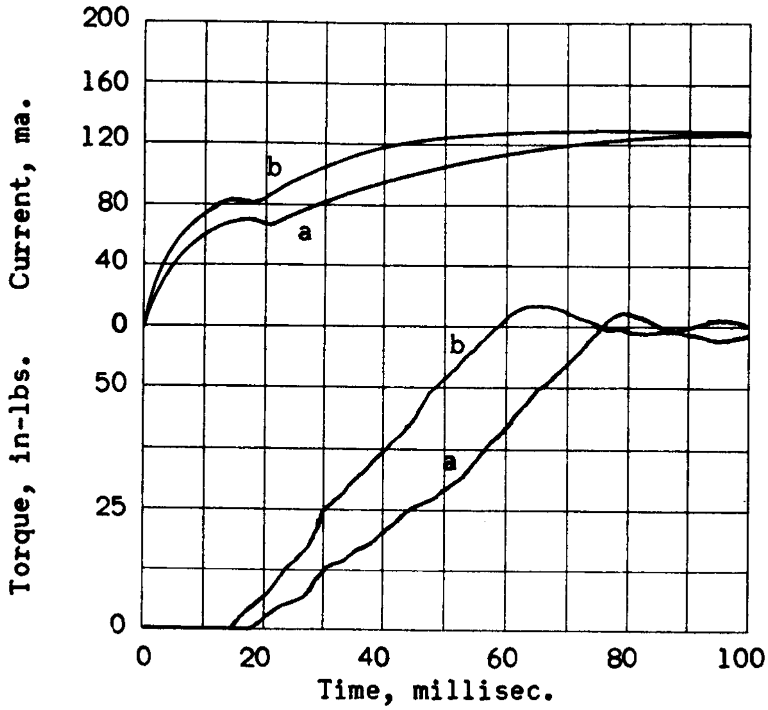


Figure 32. Effect of a Forcing Circuit.

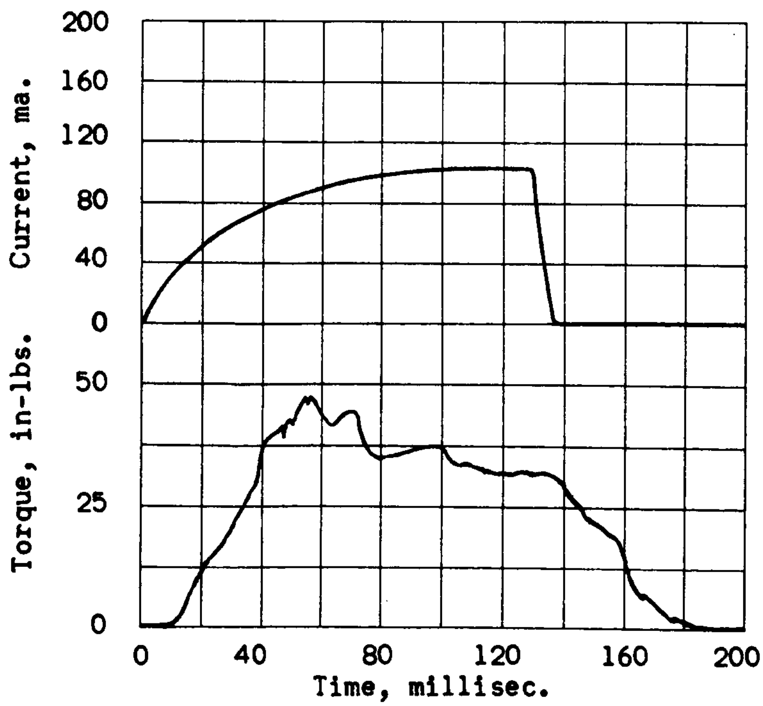


Figure 33. Histories of Clutch B.

higher than the rated torque of Clutch B. Torque was allowed to build up to its maximum (limited by the pressure between the plates), and the clutch was then disengaged.

Figure 35 shows the effect of changing the gap across which the discs must move to make contact. Curve "a" was the run in which the smaller gap was used.

Data for another disc clutch, Clutch C are shown in Figure 36 through Figure 38. Figure 36 shows the time history of current and torque, input speed 1000 r.p.m. and applied voltage the rated 12 volts. The discontinuity in the torque curve should be neglected. It was caused by the sudden arcing across the switch when it was opened. The discontinuity is more prominent for this clutch than for most of the others because current was higher. A closer look at the build-ups are shown in Figure 37, and torque decay is well illustrated in Figure 38. In both cases the input speed was 1000 r.p.m. and applied voltage was 12 volts d.c.

Build-ups for Clutch D for three different applied voltages are shown in Figure 39, again accomplished by triple exposure. Input speed was 1000 r.p.m. Applied voltages were a) 35, b) 28, and c) 20 volts d.c.

Curves for the smallest clutch tested, Clutch E, are shown in Figure 40 and Figure 41. Figure 40 is the complete time history of torque, applied voltage 24 volts d.c. (rated) and input speed 500 r.p.m. Figure 41 shows how the time histories vary when applied voltages are varied. Speed again was 500 r.p.m., and voltages were a)16, b) 20, c) 24, and d) 27 volts d.c.

Figure 42 through Figure 44 are representative data obtained for

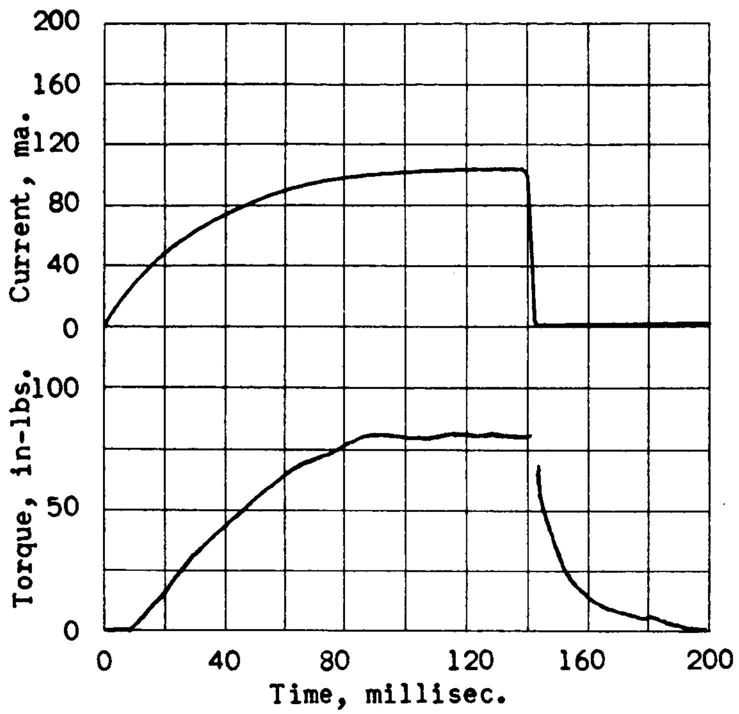


Figure 34. Maximum Torque of Clutch B.

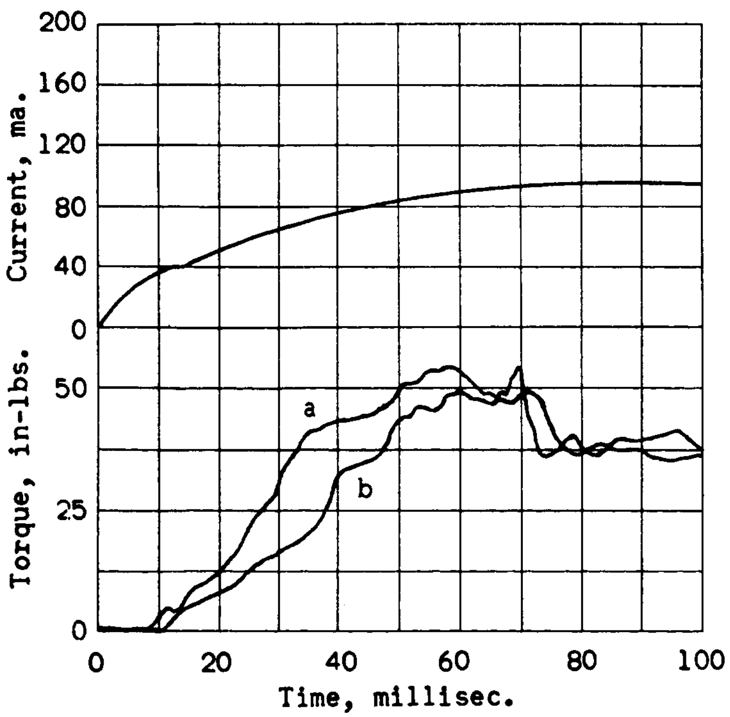


Figure 35. Effect of Gap Between Plates.

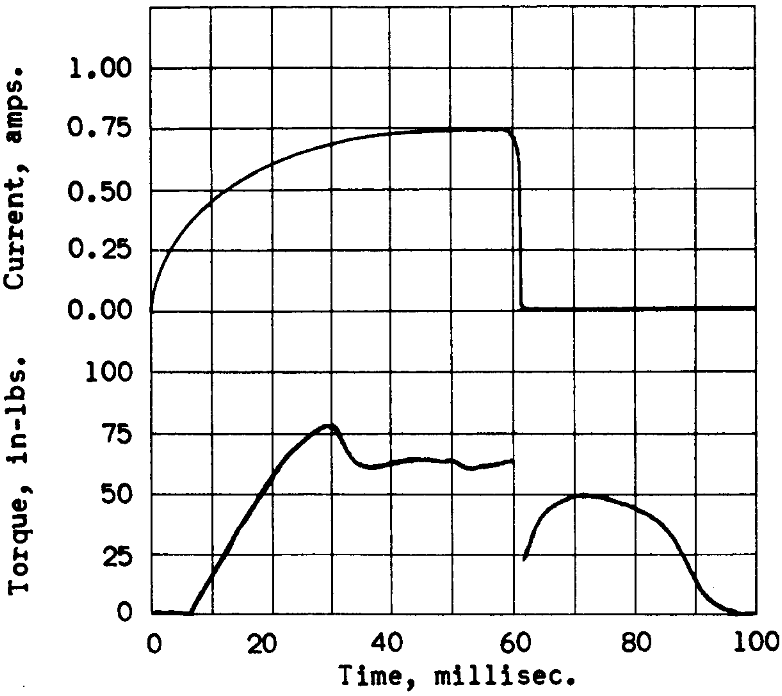


Figure 36. Histories of Clutch C.

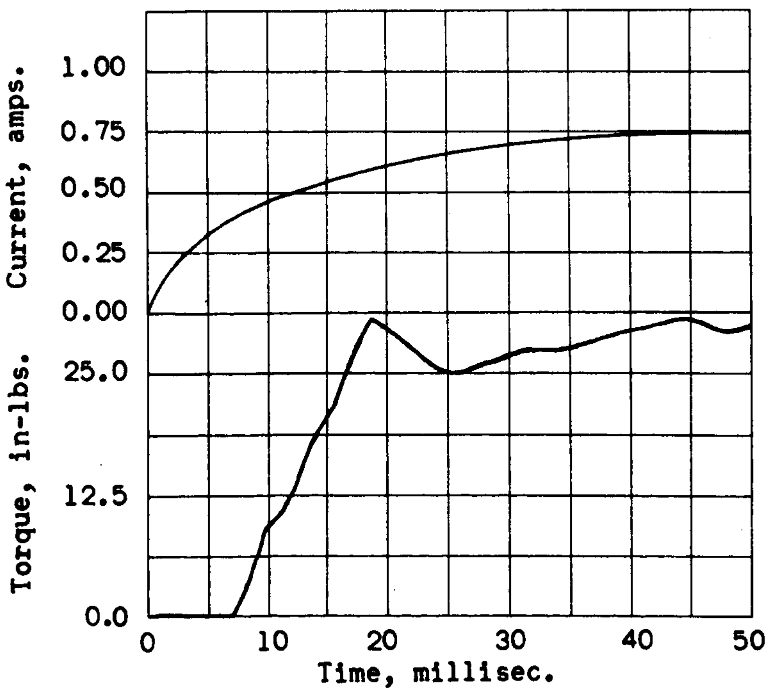


Figure 37. Build-ups of Clutch C.

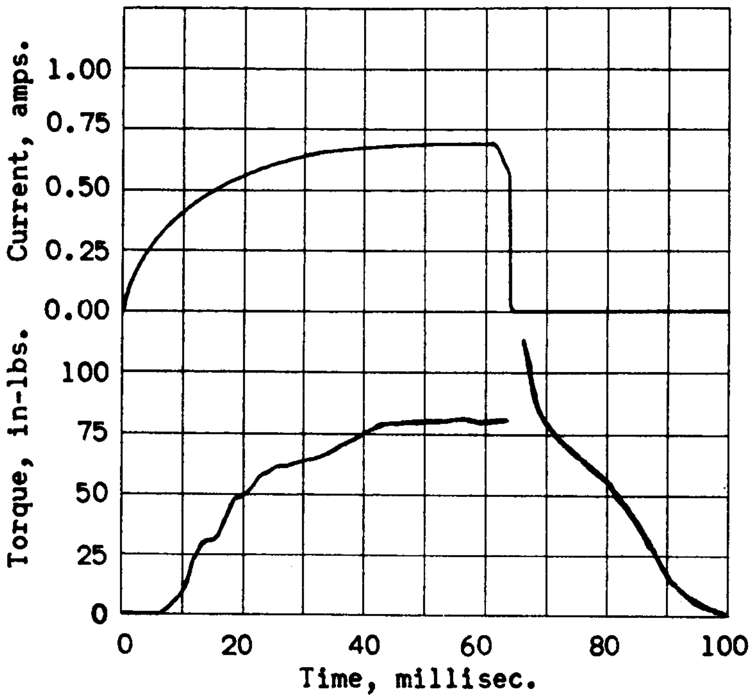


Figure 38. Histories of Clutch C.

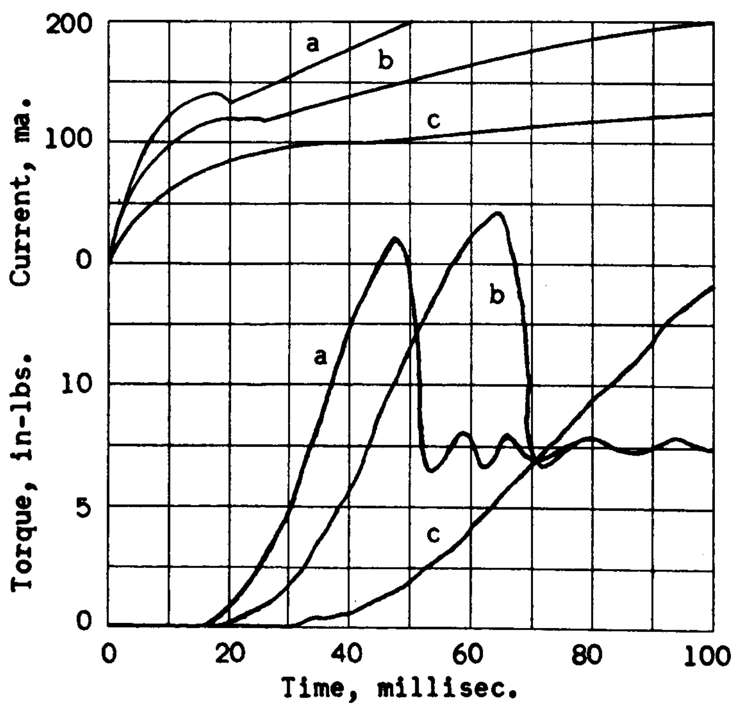


Figure 39. Variation of Build-ups with Voltage of Clutch D.

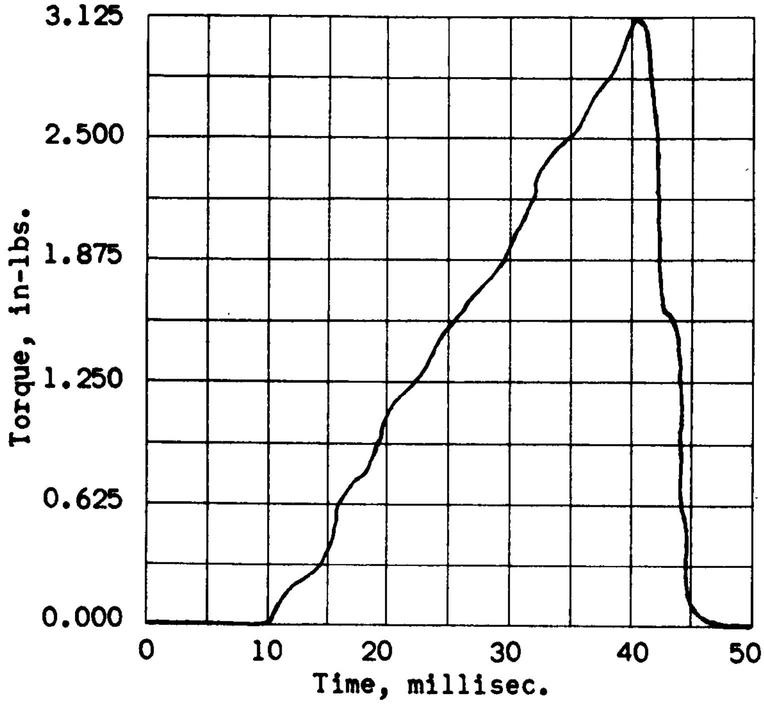


Figure 40. Torque History of Clutch E.

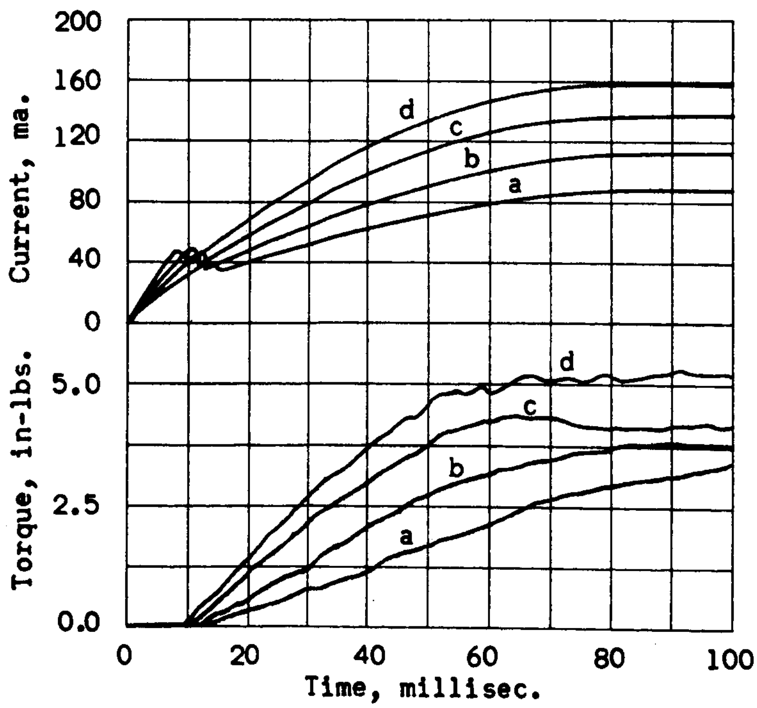


Figure 41. Variation of Build-ups with Voltage of Clutch E.

the pneumatically actuated clutches that were tested. Figures 42 and 43 are of Clutch F, the former a complete time history of one cycle. Figure 43 shows how this history changes when supply pressure is varied. In each run, the supply pressure was what the pressure curve approached, to wit, 80, 60, and 40 psi. Figure 44 shows a complete time history for Clutch G, speed 1000 r.p.m. and maximum pressure 70 psi.

Figures 45, 46, and 47 show data obtained from tests on spring clutches. Figure 45, shows current and torque build-ups for Clutch H. A 24 volt d.c. solenoid supplied by the manufacturer actuated the clutch.

Time histories of torque and current for Clutch I are presented in Figure 46 showing the effect of increased voltage on the response. Curve "a" is at rated voltage, 24 volts d.c., and the curve "b" is for an applied voltage of 30 volts.

Figure 47 illustrates the torque build-up of Clutch J. The solenoid used on Clutch I was not strong enough to actuate Clutch J, so a 115 volt a.c. solenoid was used.

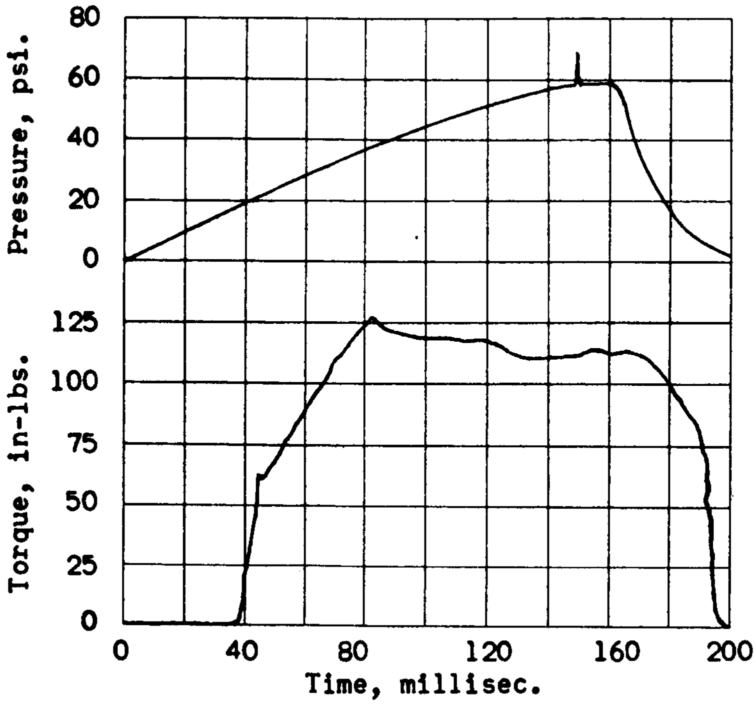


Figure 42. Histories of Clutch F.

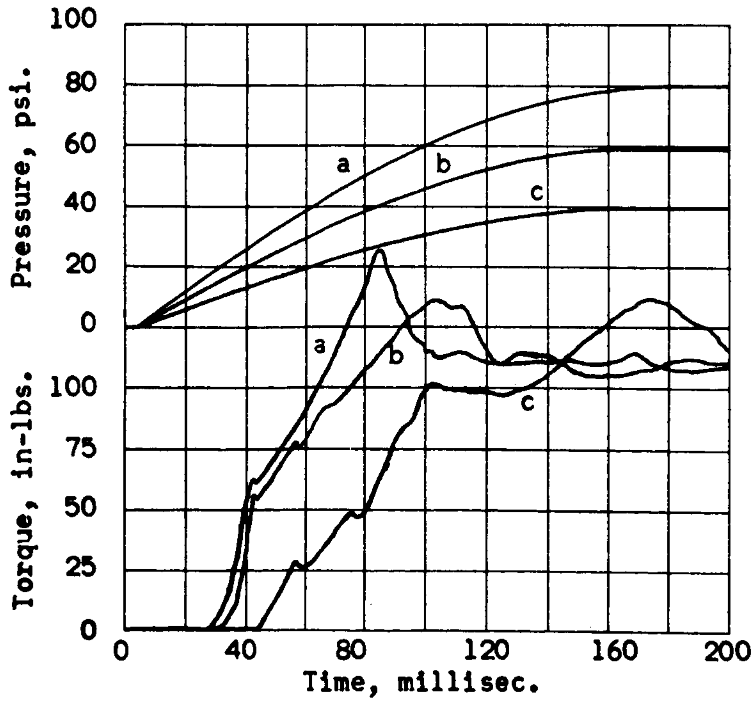


Figure 43. Variation of Build-ups with Pressure of Clutch F.

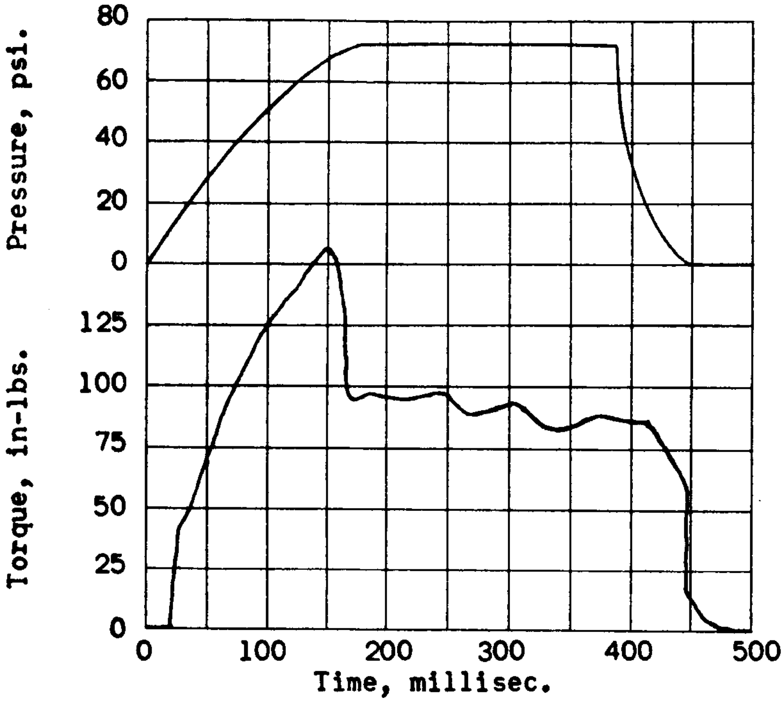


Figure 44. Histories of Clutch G.

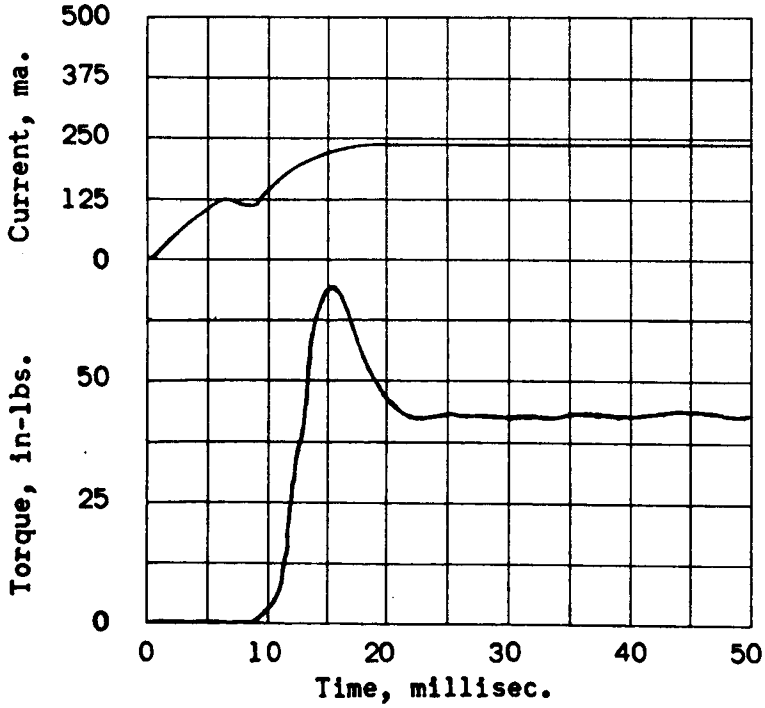


Figure 45. Build-ups of Clutch H.

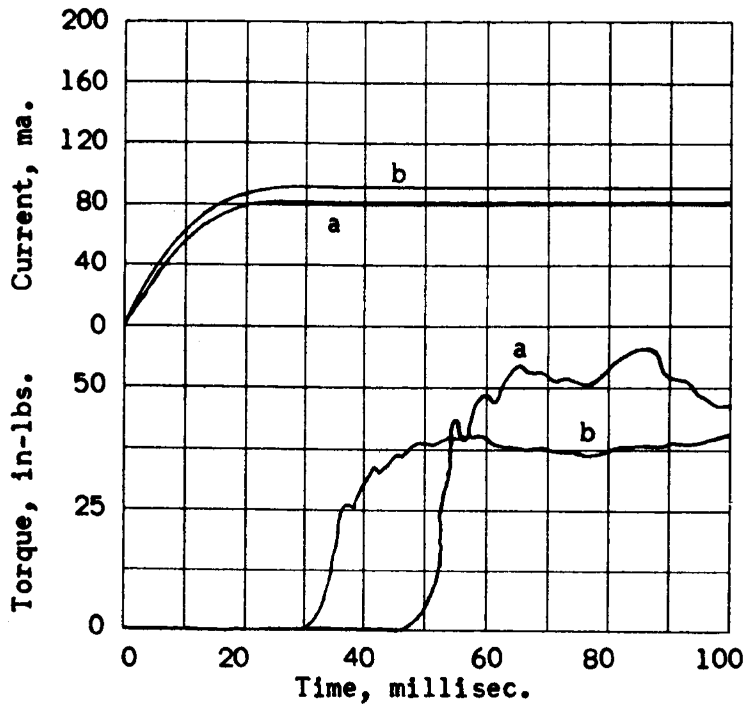


Figure 46. Histories of Clutch I.

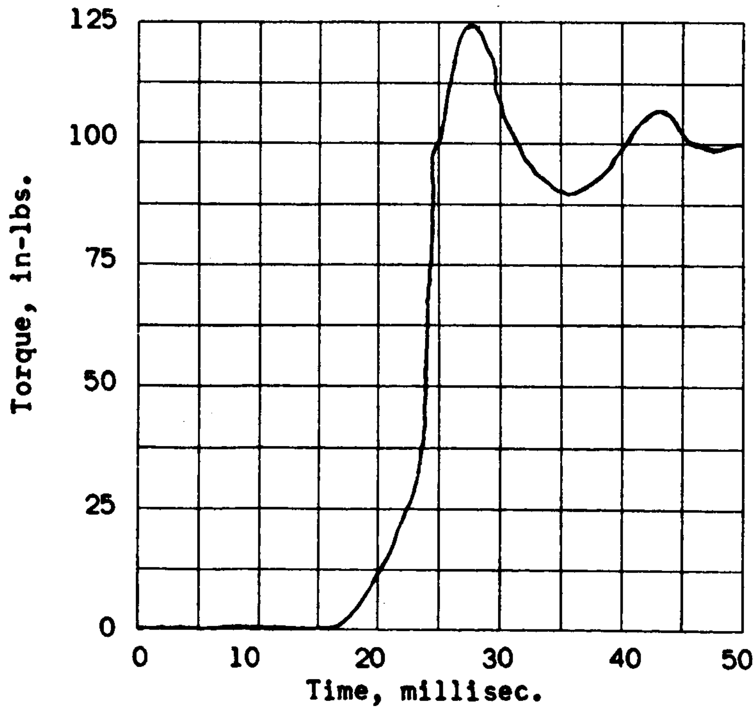


Figure 47. Torque Build-up of Clutch J.

APPENDIX C

ERROR ANALYSIS

Consider Figure 48, which represents the test apparatus from the clutch output, through a shaft with torsional stiffness, k_1, of 37,000 inch-pounds per radian and the transducer with torsional stiffness, k_2, of 30,000 inch-pounds per radian, to the fixed end. I_1 is an equivalent inertia for the worst test case, including the inertia of the output of the clutch, couplings and shafts up to the center of the transducer. I_2 is an equivalent of all the inertia from the center of the transducer to the fixed end. I_1 and I_2 were found to be 4.4×10^{-3} and 0.7×10^{-3} in-lbs-sec.^{2} respectfully. Damping has been neglected.

Figure 48 can be represented by the matrix equation

$$\begin{vmatrix} I_1 s^2 + k_1 & -k_1 \\ -k_1 & I_2 s^2 + k_1 + k_2 \end{vmatrix} \begin{vmatrix} \theta_1 \\ \theta_2 \end{vmatrix} = \begin{vmatrix} T \\ 0 \end{vmatrix} \qquad \text{(C.1)}$$

where s is the Laplace operator. Solving for $\theta_2(s)$,

$$\theta_2(s) = \frac{\frac{k_1}{I_1 I_2} T(s)}{s^4 + s^2 \frac{(I_2 k_1 + I_1(k_1 + k_2))}{I_1 I_2} + \frac{k_1 k_2}{I_1 I_2}} \qquad \text{(C.2)}$$

Substituting the values of I_1, I_2, k_1, and k_2 into equation (C.2),

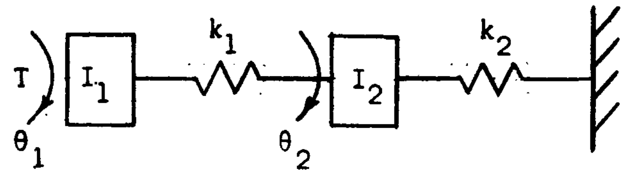


Figure 48. Model for Error Analysis.

$$\theta_2(s) = \frac{12 \times 10^9\ T(s)}{s^4 + s^2(104 \times 10^6) + 360 \times 10^{12}} \qquad (C.3)$$

Assuming the torque to be a ramp function, as was the case during testing, i.e.

$$T(s) = \frac{1}{s^2}$$

then

$$\theta_2(s) = \frac{12 \times 10^9}{s^2(s^2 + 100.5 \times 10^6)(s^2 + 3.58 \times 10^6)} \qquad (C.4)$$

Transforming back to the time domain,

$$\theta_2(t) = 33.3 \times 10^{-6}t - 1.8 \times 10^{-8}\sin\ (1.9 \times 10^3)t + 1.2 \times 10^{-10}\ \sin(10^4)t \qquad (C.5)$$

It is important to note that the coefficient of the first sinusoidal term is almost 1/2000 of the coefficient of the first term, and the coefficient of the second sinusoidal term is less than 1/270,000 of the coefficient of the first term and, therefore, they introduce no significant error.

The frequencies of the sinusoidal terms are 302 cps and 1600 cps. The filter between the preamp and the scope attenuated frequencies above 50 cps, thus reducing even further the amplitude of any signal due to the natural frequencies of the test apparatus.

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Table 5. Key: Clutch Manufacturers

Clutch	Manufacturer	Model No.
A	Fawick Corporation	SC-275
B	Stearns Electric Corp.	3.5 SMR
C	American Precision	LL-CB-40
D	Autotronics, Inc.	C-12-3
E	Reeves	SR-3066-4
F	Conway Clutch Co.	858-3E
G	Horton Manufacturing Co.	Air Champ Flywgt.
H	Precision Specialties, Inc.	SA-500
I	Precision Specialties, Inc.	EMC-1
J	Marquette Div., Curtis-Wright Corp.	C-93-5